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# Design of A Fuel Cell Moisture And Energy Recovery System

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The objective of this project was to design and develop a prototype energy and moisture recovery system for air base fuel cell system. The primary tasks in the project were the design, development, and preliminary testing of the prototype energy and moisture recovery system. The deliverable items are a final report and a prototype energy and moisture recovery system. This report is the deliverable document.

In the project, both heat exchanger and turbocharger based systems were considered. A turbocharger based system was selected. With a heat exchanger the only available heat sinks are incoming air, water, and fuel. Using these fluids limits the exhaust temperature to a value somewhat above ambient temperature. This constraint would limit moisture recovery severely, especially in warm weather. Since the fuel cell under consideration operates at elevated pressure, it is possible to use a turbocharger to cool and condense the moisture in the reaction products. Preliminary simulations and testing reveal that much of the moisture in the products of reaction can be condensed and recovered. The turbine also extracts power from the hot gases to compress the reaction air. Condensed moisture in the turbine exhaust is removed with a commercial water separator.

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## Executive Summary

The objective of this project was to design and develop a prototype energy and moisture recovery system for air base fuel cell system. The primary tasks in the project were the design, development, and preliminary testing of the prototype energy and moisture recovery system. The deliverable items are a final report and a prototype energy and moisture recovery system. This report is the deliverable document.

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# FINAL REPORT
# OF PROJECT ENTITLED

# DESIGN OF A FUEL CELL MOISTURE AND ENERGY RECOVERY SYSTEM

**PREPARED FOR**

**U. S. AIR FORCE, TYNDALL AFB**
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# CHAPTER I

## INTRODUCTION

The objective of this project is to design and develop a prototype energy and moisture recovery system for air base fuel cell system. The project is identified with Georgia Tech project numbers E25-6A1 and 25066A1 and is identified by the Air Force as NSN AZ14-98-213-2000 and Purchase Request Number F5ESCW023005. Work on the project actually began after contract approval on 1 November 2000.

The primary tasks in the project were the design, development, and preliminary testing of the prototype energy and moisture recovery system. A general view of the system is shown in Figure 1-1. The deliverable items from the initial phase are a final report and a prototype energy and moisture recovery system. This report is the deliverable document.

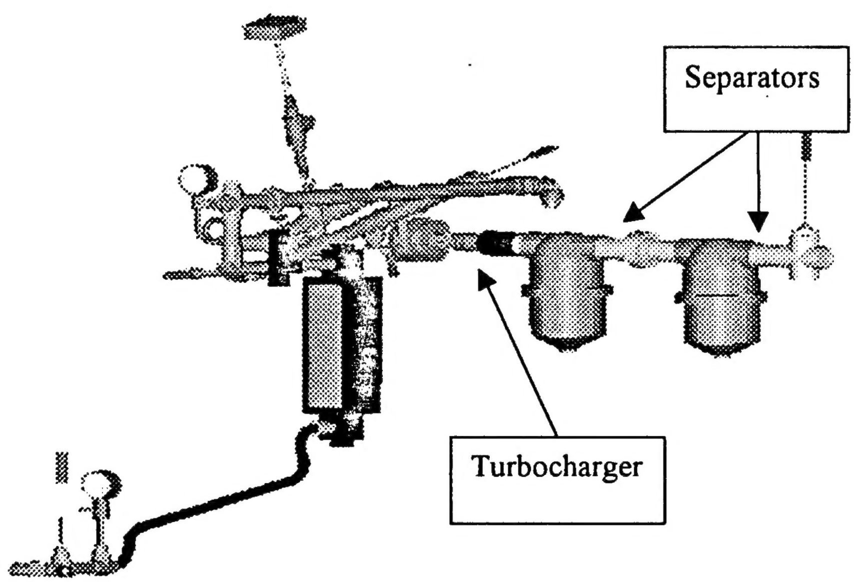


**Figure 1-1. Overall view of the test bed**

**and the prototype energy and moisture recovery system.**

In the project, both heat exchanger and turbocharger based systems were considered. A turbocharger based system was selected. With a heat exchanger based system the only available heat sinks are incoming air, water, and fuel. The incoming air is by far the largest available heat sink. Using the incoming air and fluids limits the exhaust temperature to a value somewhat above ambient temperature. This constraint would limit moisture recovery severely, especially in warm weather. Fortunately, the fuel cell under consideration operates at elevated pressure; therefore, it was possible to consider a turbocharger to cool and condense the moisture in the reaction products. Preliminary simulations and testing reveal that much of the moisture in the products of reaction can be condensed and recovered. The turbine also extracts power from the hot gases, which is used to compress the reaction air supplied to the fuel cell. Condensed moisture in the turbine exhaust is removed with a commercial water separator.

The operational conditions for the turbocharger and separator in this application are somewhat unusual. In particular, the turbine is usually expected to operate with much hotter air and not to handle condensing moisture. Even though detailed performance data for the turbocharger and the moisture separator were not available, a system that appears to be efficient and effective was designed. The turbocharger and the separator are the only especially critical components, and both have been tested and are commercially available. No high-risk components requiring special design or testing were identified.

A test bed that supplied properly heated and humidified pressurized air to the energy and moisture recovery system was constructed at Georgia Tech. Preliminary tests of the turbocharger and moisture separator have been completed. Both components and the overall system appear to perform well. No significant problems have been encountered, but the usual preliminary shake down testing of the system and calibration of the instrumentation must be completed before reliable quantitative results can be obtained.

The preliminary tests were completed before the scheduled end of work on the project on 30 September. Note that since work on the project began one month late, the work was actually finished one month ahead of schedule. The components of the prototype system that will be needed by the Air Force are the turbocharger and the separator. A separator has been received.

Some difficulty was encountered with ordering a turbocharger for the Air Force, but a unit equivalent to the one in the Georgia Tech system is now on hand. It will be retained at Georgia Tech for continued testing, and it will be provided to the Air Force when requested.

While it is clear that the prototype moisture and energy recovery system is effective, efficient, and reasonably compact, it is also apparent that some additional information is needed to support the design and integration of the overall air base fuel cell system. Therefore, a no cost extension to the project was requested to allow the following immediately needed tasks to be completed.

(1) Prepare a detailed bill of materials and equipment that will allow the Air Force to duplicate our test system.
(2) Prepare a written procedure for operating the test system.

Note that the original project only required Georgia Tech to provide the prototype energy and moisture recovery system, not a test bed to evaluate the prototype. This addition will help the Air Force complete its own test bed. This revision was discussed with the Air Force Technical Representative and approved. These two new deliverable items are included in this report.

Some very important results were observed only during the extended part of the project. Review of the Excel spreadsheet used to analyze the experimental results revealed a simple coding error. The analysis that indicated the experimental turbine efficiency to be much lower than expected with humid air rather was incorrect. In fact the turbine efficiency is correctly computed to be about 71 % when operated at a 2.8 to 1.0 pressure ratio. The calculated moisture recovery at this efficiency is about 16 %. A correct calculation of the moisture recovery assuming a reversible turbine would only be about 26 %. Even the reversible performance is much less than the approximate 65 % moisture recovery necessary to satisfy the stoichiometric water balance. One compensation is that the exhaust temperature is relatively high. Consequently, an alternative design employing either an upstream heat exchanger or a downstream combined heat exchanger and separator should be viable.

Clearly the next stage of the project should include the development of a condenser or possibly a combined separator and condenser to recover the additional water needed to balance the water requirement of the system.

It is also clear that some further information is needed that will require additional testing and modeling. The further information that is required is in two categories. First, quantitative performance data over the widest feasible range of conditions for both the turbocharger and the moisture separator and condenser are necessary. These data are necessary to quantify the sensitivity of the moisture and energy recovery system to variations in the operating conditions. Second, a appropriately detailed and validated model of the energy and moisture recovery subsystem is necessary to support the overall system modeling, integration, and optimization. The appropriate level of detail is an integrated subsystem model that includes engineering models for the critical components. These component models should be on at the level of finite thermodynamic control volumes, not detailed continuum computational fluid dynamics models. The critical components are the turbine, compressor, separator, and condenser. This subsystem model can be exercised independently or integrated with a model for the balance of the fuel cell system to support design studies. This information should be obtained during a proposed second phase of this project. At this point sufficient information should be available to support an efficient and effective overall design.

## CHAPTER II

## NARRATIVE

The project was administratively initiated on 1 November 2000. The project initiation meeting was held at Georgia Tech on 2 November 2000. The project definition and schedule were discussed. It was understood that the testing of the prototype system at Georgia Tech would be limited to shake down testing not comprehensive testing and evaluation. It was further understood that the prototype system would be delivered to the Air Force and that the Air Force would complete the comprehensive testing and evaluation.

For the first month of the project, the schedule called for definition and conceptual design, and during that time we concentrated on these tasks. Conceptual designs were developed that included turbocharger based systems, heat exchanger based systems, and combinations. The technical and commercial literature and data sources were searched for performance and design information on turbochargers. Technical literature on moisture separators for use with turbocharger systems was also searched. Various heat exchanger designs were considered. Cross flow, counter flow, and heat pipe based heat exchanger systems were all considered.

Several project-definition teaks were also started during the first month. A spreadsheet of properties at critical locations in the system was received from AFRL/MLQD. This information was invaluable in generating an appropriate system design. Information on practical considerations in PEM fuel cell systems were searched and evaluated. Information on commercial turbochargers and turbocharger applications was searched. Finding helpful engineering information on small turbochargers was a challenge throughout the project. Information on air cycle cooling systems, which are analogous to this application was also reviewed. Even though all air systems are common, especially in aircraft, finding helpful engineering information these systems was also a challenge throughout the project

Immediate tasks were to develop a simple psychometric model for recovery of moisture and energy from the fuel cell exhaust. The promising conceptual designs were incorporated into this model to evaluate the conceptual design alternatives.

The second month was one of the most important in the project. During this month the conceptual design was successfully completed, and the preliminary design was begun. The turbocharger system was found to be the best choice for the preliminary design. The heat pipe heat exchanger design was still available as a back up with low technical risk but low limiting performance. No long lead-time items, such as a larger boiler, were identified as being necessary. At this point in the project, it was projected that it should be possible to finish the preliminary design in the next, the third, month of the project; but some high priority tasks existed before a preliminary design with limited technical risk and high probability of success could be completed. The main challenge was that sufficient information on the performance of the small turbines used in turbochargers had not been found. This problem continued throughout the project. The literature search could not of course be exhaustive, but it was extensive. Even so, no really useful practical performance data was ever found in the open literature. The literature survey was continued, and more manufacturers were contacted in an effort to find suitable smaller turbines and performance data on smaller turbines in general. Further analysis of the turbine expansion process when handling humid air was still necessary and was continued. At this point, the highest priority was finding information from the technical literature or commercial sources so that the technical risk could be minimized. Because of the small size of the turbocharger needed and the unusual operating conditions, no complete and directly useful information was ever found.

In the third month, we continued to have confidence in the primary conceptual design, which was the turbocharger based system, and the back up design, which was a heat exchanger based system. These designs had been compared at a conceptual design level, and the evaluation of the designs at a preliminary level continued during this month. This evaluation was slowed by difficulty in obtaining performance data from the turbine vendors on a timely basis.

The literature review to support the preliminary design and the preliminary design evaluation was continued vigorously but with limited success. Only at this point was uncovered any promising research literature on the modeling and performance of all-air refrigeration systems, which is the conventional technology most closely related to this application.

During the third month we identified some promising turbochargers and found one at least marginally suitable unit on hand at Georgia Tech. This finding showed that a turbocharger of size suitable for the prototype test was commercially available.

During the third month we began to configure the laboratory space for the experiment, and we developed the requirements for any additional utilities needed in the laboratory. Fortunately since the utility installation was ultimately delayed, it was not necessary to route additional electric power to the lab.

During the fourth month, the preliminary design was essentially completed. Specifications for utilities needed in the laboratory were also completed. The turbocharger vendor provided turbine and compressor performance maps. Also, a thermodynamic model for the irreversible expansion of moist air through a turbine was completed. This model was used to plan and evaluate the experimental work.

During the fourth month other tasks to be performed were to begin the detailed design of the test system, to begin modifying the utilities in the lab space, and to order any long lead-time equipment as soon as possible. An apparently suitable turbocharger that was ultimately used in the prototype was received during the fourth month. This design and procurement work continued into the fifth month. Also some preliminary work on the alternative concept, the heat pipe based condenser, was begun; but it was not found necessary to continue this work beyond the fifth month.

During the fifth month, specifications for utilities needed in the laboratory were completed, and the installation work was requested. Installation was actually delayed until the seventh month.

The final design at a schematic level sufficient to support the construction of a prototype was essentially complete during the sixth month. Utilities needed in the laboratory were installed by the start of seventh month. The initial configuration prototype test system was nearly complete by the end of the seventh month.

At that time the remaining important component to be selected was the moisture separator. At that time four alternatives were being considered:

1. centrifugal separator
2. cyclone separator
3. coalescing filter
4. impingement separator

Ultimately it was found that only a centrifugal separator was readily available in the size needed for this application.

By the start of the eighth month, all electrical power circuits were in place. An existing thermostatically controlled electrical power supply system was used to power the duct heaters. This system was modified and put into place. Then it was necessary is to run conductors from the control system to the duct heaters. The electrical power source for the boiler was in place, and the system was tested with cold air and hot dry air before the boiler was installed and energized. Then the system was tested with hot moist air. In summary, the prototype test system was mechanically complete, and all electrical support systems were on hand at the end of the eighth month.

The preliminary tests during the eighth month revealed two small problems. One was a subtle problem with the turbocharger oil system. The original oil system did not follow standard automotive practice. Actually this practice is not well documented but merely understood within the automobile industry. The original oil return line did not slope continuously to the oil sump, so the bearing housing was inadvertently flooded with oil. This situation caused excessive

leakage of oil into the turbine. In consequence, the turbine exhaust carried a significant mist of oil droplets. This problem was corrected by properly routing the oil return line. It is actually necessary that the oil return line slope continuously to the oil sump. The flow in this line should only partially fill the tubing. The oil line and the oil sump were redesigned and reworked so that oil now returns freely to the oil sump. This modification solved the oil leakage problem.

The second problem had been sealing the duct heaters. The original heater ducts were mere thin walled aluminum tubes with flimsy sheet aluminum flanges. Consequently, these aluminum flanges were so flimsy and uneven that they leaked even though they were connected to substantial brass pipe flanges and were sealed with silicone rubber gaskets. The original foam rubber gaskets were almost blown out of the flange joints by only moderate pressure. A harder rubber minimized this problem, but the joints still leaked somewhat. The stiffer gaskets were reinforced and retained by silicone RTV and adhesive aluminum foil duct joint tape. This modification appeared to have stopped the leakage at the joints. Some leakage still existed at the electrical penetrations, but silicone RTV was used to stop these minor leaks. These repairs and modifications appeared to solve the problems with the resistance heaters; however, the leaks reappeared and worsened as the operating pressure was increased. Ultimately it became necessary to replace the commercial heaters with special heaters designed and assembled at Georgia Tech.

The minor mechanical problems appeared to be solved, so preliminary performance testing and final installation of instrumentation and integration with the data acquisition system was begun. All instrumentation was in place, and the remaining data acquisition components were later received. Completion and testing of the instrumentation and data acquisition system were the next objectives.

Considerable progress was made during the ninth month. Structural and mechanical construction of the prototype was essentially completed, but a couple of minor problems had developed. All electrical power circuits were in place. A thermostatically controlled electrical power supply system had been modified and installed to power the duct heaters. The system had been tested with hot dry air, and the instrumentation and controls worked well. The main

problem that developed was that but the duct heaters were found to leak excessively at moderate pressure. As described above, some considerable effort was made to develop a better seal for the heaters. This effort ultimately failed, and a replacement heater system was designed. The components for the new heater were ordered. Most testing could proceed even without the new heater with a pipe in the place of the heater. The electrical power source for the boiler was in place, but the system was further tested with cold air and hot dry air before the boiler was installed and energized. One instrumentation problem was discovered. The recommendations from the rotameter supplier to modify the rotameter currently in use were somewhat misleading. After installing the new rotameter, it is found to be off scale at the relatively low pressures where the turbine actually operates. Ultimately the piping system was revised so that the pressure was higher at the rotameter, and the local volumetric flow at the rotameter was kept small enough to be measured. In summary, at the end of the ninth month the prototype test system was mechanically complete except for the new heater.

At the end of the ninth month, the minor mechanical problems appeared to be solved or on the way to resolution. Most of the necessary instrumentation was in place, and it was possible to return to work on the preliminary performance testing, final installation of instrumentation, and integration with the data acquisition system. Installation of the boiler was the next objective.

The one remaining functional component previously not selected was the moisture separator. A qualitative comparison revealed that a centrifugal separator would be the best choice. No suitable axial separator was ever found for this size unit, but a suitable vortex flow centrifugal flow separator was found and ordered.

By the end of the tenth month, structural and mechanical construction of the prototype was essentially complete. The boiler and the vortex flow centrifugal separator had been installed and tested. A couple of minor problems that were reported earlier had been resolved. A leak proof heater that was designed and assembled here replaced the leaking duct heaters that were originally purchased. Operating the rotameter at a somewhat higher pressure and allowing the pressure to drop after the rotameter resolved the rotameter range problem. At higher pressure the actual volumetric flow is lower for a given standard flow. The new arrangement keeps the

volumetric flow low enough for the range of the rotameter. In summary, the prototype test system was mechanically complete and has been tested at the end of the tenth month. The system appeared to operate well driving the compressor and removing a considerable amount of liquid water from the flow.

The next tasks to be completed were to install the remaining instrumentation, to accomplish the necessary calibration of instrumentation, to program and interface the data acquisition computer and software, and to begin performance testing of the turbocharger based system.

The last scheduled month of the project was the eleventh month. At this time, the prototype test system was mechanically complete, and it had been instrumented and tested. The system appeared to operate well driving the compressor and removing a considerable amount of liquid water from the flow. Preliminary performance testing was completed. Some contamination of the lubricating oil with water was noticed. Presumably this water originates in the condensing steam. This is a potential problem that should be addressed before this particular unit is recommended for field applications. This problem will be discussed with the manufacturer. One potential solution is to increase the oil pressure, and this approach should eventually be tested.

Preliminary testing during the tenth month erroneously indicated the turbine efficiency to be much lower than expected with humid air rather in comparison with dry air. Considerable time and effort was actually wasted attempting to understand this problem. The actual problem was a minor coding error in the Excel spreadsheet used to analyze the test data.

Some very important results were actually observed in the eleventh month. Review of the Excel spreadsheet used to analyze the experimental results revealed a simple coding error. The analysis that indicated the experimental turbine efficiency to be much lower than expected with humid air rather was incorrect. In fact the turbine efficiency is correctly computed to be about 71 % when operated at a 2.8 to 1.0 pressure ratio. The calculated moisture recovery at this efficiency is about 16 %. A correct calculation of the moisture recovery assuming a reversible turbine would only be about 26 %. Even the reversible performance is much less than the

approximate 65 % moisture recovery necessary to satisfy the stoichiometric water balance. One compensation is that the exhaust temperature is relatively high. Consequently, an alternative design employing either an upstream heat exchanger or a downstream combined heat exchanger and separator should be viable.

At this point the originally proposed work has actually been completed as a prototype moisture recovery system has been designed and has been given a preliminary test. If it is possible to continue with this work, the next tests should try to confirm the turbine performance with enhanced instrumentation including measurement of the shaft speed and measurement of the compressor work. Once the turbine performance is confirmed, we hope to begin work on the alternative heat exchanger or cooled separator designs. The development of a list of materials and a test procedure, which are the additional deliverables required when the project was extended, have been completed. This report represents the final deliverable and the completion of this project.

# CHAPTER III

# BACKGROUND

This chapter presents background information about fuel cells (FCs) and related technologies that have motivated this research. The scope of this report is to develop technology for extracting moisture and energy from the exhaust stream of a Proton Exchange Membrane (PEM) FC operated at a pressure above ambient. In particular, this chapter will provide information about fuel cells and gas-liquid separation. General FC theory, as well as in depth analysis of the PEM type FCs, will be discussed. An overview of how a FC operates will be provided to illustrate the advantage of operating at higher pressures and temperatures. Information on FC operating conditions is useful in this experiment as the exhaust stream conditions are directly dependent upon the operating conditions. Gas-liquid separator performance will also be discussed in this section.

## III.1 Fuel Cell Fundamentals

Fuel cells provide a means of converting the chemical energy of a chemical reaction into more easily usable electrical energy. There are many types of fuel cells, but all operate in generally the same way. This section provides a background for understanding generic fuel cell operation and function.

### III.1.a General Fuel Cell Fundamentals

FCs consists of two electrodes, a cathode, or positive electrode, and an anode, or negative electrode. The two electrodes are separated by an electrolyte. Figure 3-1 (Brown *et al*, 1999) shows a general fuel cell. An electrolyte is a conductor through which electric current is carried by ions instead of free electrons. Fuel, typically gaseous hydrogen, is fed into the anode side, where the electrode encourages the splitting of the hydrogen atoms into electrons and protons. The protons pass through the electrolyte and into the cathode. The electrons are conducted

through an external circuit, where the electric energy is extracted. The cathode is continuously fed with an oxidant, usually air, which combines with the protons and electrons to form water.

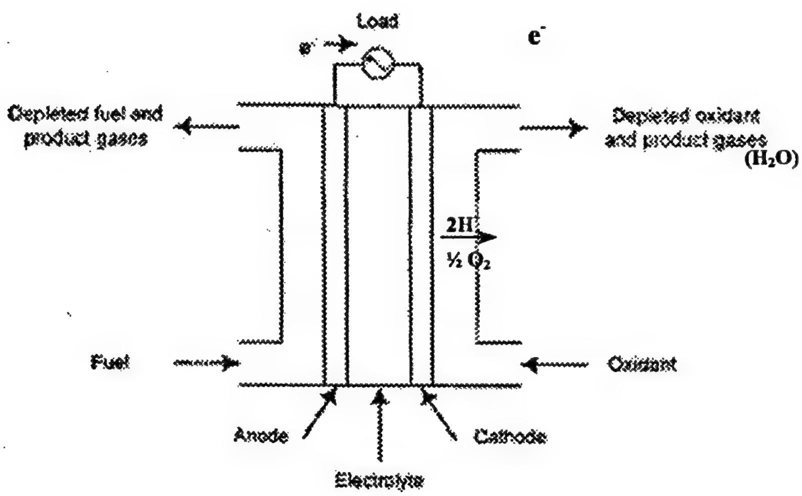


**Figure 3-1 General fuel cell schematic.**

This section has provided general components and operation of a single generic FC. Fuel cell systems are comprised of stacks. Each stack has a set power output based on its size and operating conditions. Many stacks are combined in a FC system to achieve the desired total system power output.

III.1.b Proton Exchange Membrane Fuel Cells

There are primarily four types of FCs available at the present time: Proton Exchange Membrane (PEM), Phosphoric Acid (PAFC), Molten Carbonate (MAFC), and Solid Oxide (SOFC). PEM FCs have a polymer ion exchange membrane electrolyte and operate at approximately 80 C (*ca.* 180 F). A PAFC operates at approximately 200 C (*ca.* 400 F) and has an electrolyte composed of liquid phosphoric acid. The low operating temperature of the PEM and PAFC type FCs make internal reformation impossible; and therefore, both require hydrogen fuel be supplied directly to them. The MAFC has a liquid molten carbonate electrolyte, and operates at a temperature of approximately 700 C (*ca.* 1300 F). The SOFC has a metal oxide

electrolyte and operates at temperatures in the 800 to 1000 C (*ca.* 1500 to 1800 F) range. The high temperature of the MAFC and SOFC allow the fuel to be reformed internally. Internal reformation means the carbonaceous fuel enters the FC and is combined with water internally to produce hydrogen. An internally reforming FC can accept many different fuels. The most common fuels used in high temperature FCs are carbon monoxide (CO) and methane ($CH_4$).

A PEM FC has many advantages over other FC types. PEM FCs have a polymer electrolyte. This design eliminates the risk of corrosive fluids escaping from the cell. However, water must be supplied to hydrate the membrane. The low operating temperature of PEM FCs is also attractive, as steady state conditions can be reached much faster than in any other presently available type. Low temperatures also mean the cell is safer to operate. For these reasons the PEM type FC is the type selected for remote and mobile applications and is therefore modeled in this experiment.

One other important characteristic of FCs is the chemical reactions inside the cell. Each type of FC has unique anode and cathode reactions. The anode and cathode reactions for a PEM type FC are given in equations 3-1 and 3-2 respectively. The anode or fuel side reaction is

$$H_2 \rightarrow 2H^+ + 2e^- \tag{3-1}$$

The cathode or oxidizer side reaction is

$$1/2\ O_2 + 2H^+ + 2e^- \rightarrow H_2O \tag{3-2}$$

Performing a chemical balance on the entire fuel cell yields Equation 3-3

$$1/2\ O_2 + H_2 \rightarrow H_2O \tag{3-3}$$

Overall the FC consumes hydrogen and produces water.

PEM FCs are very attractive for many applications. They appear to be the safest choice mainly due to their low operating temperature and relatively benign electrolyte. However, there are some negative aspects to PEM technology. PEM membrane hydration is a major issue. If the membrane is not properly hydrated the cell efficiency drops considerably. Therefore, water supply is important for PEM FC operation. The FC purposed for the application described in this report uses a proprietary internal hydration system: therefore, water presumably does not need to be recovered merely for hydration. Instead, the system investigated in this research will recover water for fuel reforming, which is always necessary.

## III.2 PEM FC Operating Conditions

This section discusses the typical operating conditions of a PEM FC and the effect these conditions have on the FC performance. The parameters to be discussed are temperature, pressure, fuel utilization, excess air, heat removal, and FC system water management. Water management is very important to this particular experiment in that moisture is being removed from the exhaust stream and being used in the fuel reformation process.

### III.2.a Fuel Cell Temperature

FC temperature is an extremely important issue in FC design and operation. Increase in FC temperature decreases resistance within the cell. This occurs mainly in the membrane, allowing the chemical reactions to take place at a higher rate. For example, an experiment (J.C. Amphlett *et al.*, 1992) on the effect of temperature on FC performance shows an increase in voltage output as FC operating temperature is increased. This data is summarized in Figure 3-2. The results show an increase in cell voltage of about 2 mV for each degree C increase in overall fuel cell temperature. Though FC voltage increases linearly with temperature, there is an upper limit on FC temperature. Due to the need to maintain adequate membrane hydration, FC temperatures are limited to approximately the saturation temperature of water at the pressure of FC operation. Nominal PEM FC temperatures are in the range of 70 to 90 C (*ca.*160 to 190 F), making it the lowest temperature FC.

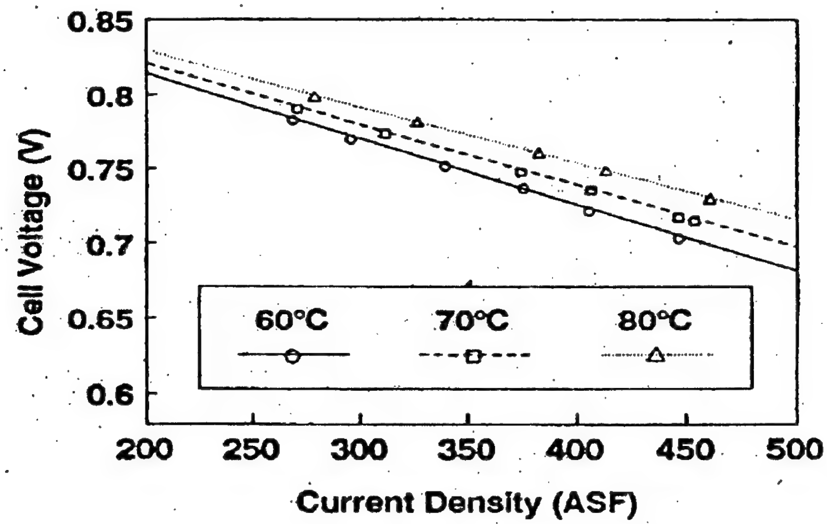


**Figure 3-2 Temperature influence on PEM FC voltage.**

FC operating temperature is very important in the experiment described in this report. Higher FC operating temperatures result in more available energy in the FC exhaust from which energy can be recovered and used to compress air

III.2.b Fuel Cell Pressure

FC pressure has an even greater effect on FC power output than temperature. FC systems running at ambient pressure have been extensively tested, and this technology is advanced and well documented.

However, pressurization of the FC results in increased FC performance. This also allows for smaller, more lightweight, systems that yield the same power as the larger units operating at a lower pressure. Research in this area shows a significant increase in power production with increasing pressure. Figure 3-3 summarizes the results of one such experiment performed in this area (J.C. Amphlett *et al.*, 1992).

FC pressure performance gains are limited only by economic factors: mainly the cost of compressing the inlet air. Typical fuel cell pressure is 0.3 MPa (*ca.* 40 psig) nominally. A typical FC is operated in the 0.1 to 1 MPa (*ca.* 15 to 150 psig) range. The FC structures

available today can withstand pressure up to 20 MPa (3000 psig) (Hirschenhoffer *et al.*, 1998). Therefore, ways to efficiently compress air without adding additional loads are very attractive to FC technology.

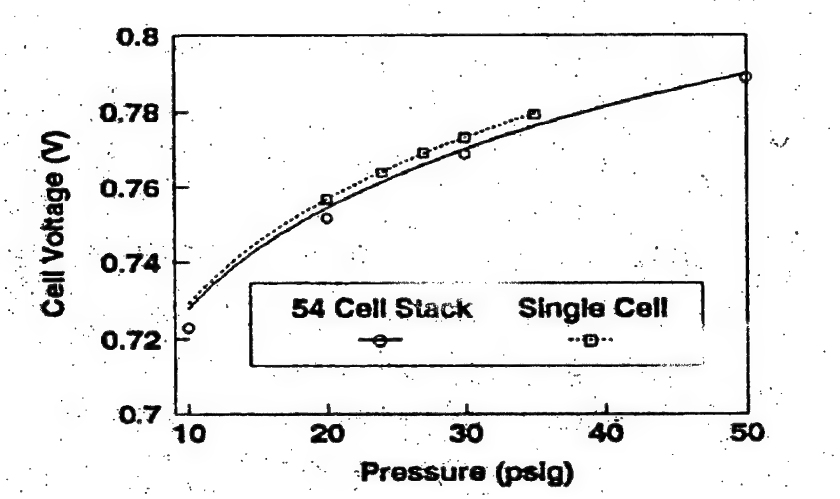


**Figure 3-3 Effect of pressure on FC voltage in a single cell and 54-cell stack.**

III.2.c FC Excess Fuel and Air

Fuel cells convert the energy of a chemical reaction into electric power. In such chemical reactions, there is typically a fuel and an oxidizer. In the case of a FC the fuel is hydrogen, and the reactant is typically air. There is a theoretical amount or air that is needed to fully react with a given amount of fuel. Almost always more air is put into the system than can react and generate heat. In combustion processes excess air may be beneficial for complete combustion. The typical range of excess air in natural gas combustion systems is around 5 to 10 percent. In FC technology this does not seem to be the case. An example study showed no increase in cell voltage for a range of 150 to 250% excess air. The same study also shows no increase in cell voltage for 100 to 125% excess fuel (J.C. Amphlett *et al*, 1992).

Fuel cells are typically operated with approximately 150% excess air (Hirschenhoffer *et al.* 1998). This excess air, as stated previously, has little direct effect on the FC combustion

process. In FC technology, excess air put into the system helps to remove excess process water from the cathode side of the FC. The removal of excess water helps to prevent membrane flooding. Membrane flooding greatly reduces the capability of the FC to produce power.

### III.2.d FC Heat Removal

PEM FC heat removal is a major issue in fuel cell design and application. PEM fuel cells typically do not exhaust a significant amount of energy; therefore, the amount of heat removal is very closely related to the FC efficiency. A common PEM FC used in automobiles, which produces approximately 50 kW and having an efficiency of 35%, will reject approximately 75 kW (M. Hagan, 2000). A larger FC has a higher efficiency; therefore, it requires less heat removal per Watt of power production. For instance, a 250 kW PEM FC has been demonstrated in Berlin and requires approximately 230 kW of heat removal (Pokojski, 1999). In addition, a model of a 250 kW PEM FC was developed and calculated the same value of 230 kW heat removal computationally (J.C. Amphlett, 1998).

Heat removal is usually accomplished through a series of air or water channels passing through the FC. Heat is transferred to the air or water and then removed from the system. A PEM FC generally rejects heat at approximately 60 to 90 C (*ca.* 140 to 190 F). The relatively low temperature of PEM FC makes the task of cooling the cell somewhat more difficult. Heat is rejected at a much lower temperature, necessitating the use of larger heat exchangers. In some instances, this heated water from the cooling system offers opportunities for cogeneration and consequently added system efficiency. For instance, the heated water from the aforementioned Berlin experiment is used to preheat the city hot water through the use of a heat exchanger.

### II.2.e FC Water Usage

PEM FC water usage is a very important issue in the scope of this project. PEM type cells require a great deal of moisture for membrane hydration and fuel reformation. Membrane hydration is critical for FC performance; therefore, the water balance is a critical FC design criteria. PEM FC technology typically uses one of two types of membrane hydration:

humidification of the inlet fuel and oxidizer or a separate system directly supplying pure water to the membrane. In addition to the water required for membrane hydration, much water is required for FC fuel reformation. To hydrate the membrane or reform fuel, moisture recovery from the exhaust stream would increase FC functionality by reducing or possibly eliminating the need for an external water supply. As stated previously the membrane hydration system for the FC examined in this investigation is proprietary and performed internally, and therefore the water recovered in this experiment will be used in the fuel reformation process.

It has been established previously that the normal PEM operating conditions are 0.1 to 1 MPa (*ca.* 15 to 150 psig) and nominally 80 C (*ca.* 180 F) operating temperature. The exhaust stream has been characterized as fully saturated (100% relative humidity) air mixed with a small amount of liquid water exiting at the system temperature and pressure. The chemical reaction within the FC actually produces more water than is required by the entire FC system. If this water could be recovered, then the water usage problems would be solved. In order to remove this water from the exhaust stream, the temperature of the exhaust must be dropped below the dew point and the condensing liquid separated from the gas. The approach used in this report is to utilize the pressure differential between the exhaust gases and ambient in order to lower the temperature of the exhaust stream. Condensate will then be removed from the air stream. This will be accomplished by the use of a turbocharger system in tandem with one or two cyclone gas-liquid separators.

## III.3 Centrifugal Separator Fundamentals

### III.3.a Gas-liquid Separation Fundamentals

This section will cover the fundamentals of techniques for separating liquid droplets from a gas-liquid stream. There are three basic types of separators used for removing liquids from a gas-liquid stream. The three types of separators are: centrifugal separators, impingement separators, and membrane separators. Each type of separator will be discussed in terms of general theory and advantages and disadvantages for liquid removal from gas streams.

The membrane type separator consists of a membrane placed in contact with a pressurized gas-liquid stream. The membrane is usually a porous or fibrous material. The denser liquid is trapped in the membrane while the less dense gas passes through the pores. The membrane is designed according to what droplet size is to be removed. If sized well, the membrane type separator typically yields good collection efficiency. However, membrane type separation results in a large pressure drop with the membrane providing a great deal of resistance. In addition, damage or fouling of the membrane may occur. This increases maintenance costs for the system and results in significant downtime.

Impingement type separation is accomplished by placing an obstacle in the flow-path of a gas-liquid stream (Setford 1995). The denser liquid particles have more momentum and collide with the obstacle. The less dense gas particles have less momentum and pass around the obstacles. The liquid drops attach to the obstacles and coalesce with other droplets forming larger droplets. The droplets increase in size until they slide down the obstacle surface and into a collection chamber. The impingement type separator like the membrane separator creates a large pressure drop. There is also a possibility of greater maintenance on the impingement type separator due to the continuous collisions between the liquid and the obstacles.

The centrifugal type separator operates by forcing the gas-liquid stream into a cylindrical vessel. The stream enters the vessel tangentially and is immediately forced to rotate around the inside of the vessel. The liquid particles are denser and therefore have more momentum. These particles have more inclination to stay in straight line than the gas particles and therefore are more prone collide with the walls of the vessel. These droplets then slide down the side of the vessel and into a collection chamber. The gas and the liquid that do not get removed initially, form a vortex moving downward within the vessel. The swirling motion and resultant centrifugal force causes more liquid to collide with the vessel wall and ultimately flow into the collection chamber. A conical section at the bottom of the separation vessel causes the vortex to reverse forcing the gas back up and out of the vessel. The force of gravity on the denser liquid droplets makes it more difficult for them to exit with the gas particles and offers a final opportunity to remove liquid. The centrifugal type separator has a small pressure drop and is very effective for

gas-liquid separation. With no moving parts, centrifugal separators require very little maintenance.

For this experiment the centrifugal separator was selected due primarily to the low-pressure drop associated with this method. The membrane and impingement type separator for the flow ranges encountered in this experiment had pressure drops too large to be considered. The centrifugal separator operates with no moving parts and thus requires very little maintenance. An off the shelf centrifugal type separator with a very small pressure drop is readily available in the flow ranges encountered in this experiment. The low-pressure drop, low maintenance, and availability, make the centrifugal separator the obvious choice for this application.

III.3.b Gas-liquid Separator Performance

The performance of a gas-liquid separator is measured in terms of particle capture efficiency and pressure drop across the apparatus. The pressure drop is primarily dependant upon the geometry of the separator, inlet velocity, and concentration of the liquid the entering gas stream. The separation efficiency is a ration of the mass of liquid in the exit stream compared with mass of liquid that enters the separator. The separation efficiency is defined in Equation 3-4.

$$\eta_{sep} = 1 - \frac{\dot{m}_o}{\dot{m}_i} \tag{3-4}$$

Where $m_o$ and $m_i$ are mass of water out and in respectively. In this experiment this calculation is possible knowing the rate of moisture entering the separator and the amount of moisture removed. From this information the outlet moisture rate can be calculated and thus the separation efficiency of the separator.

# CHAPTER IV

# LITERATURE REVIEW

This chapter summarizes the literature reviewed in this investigation. Literature was reviewed to get an idea of basic Fuel Cell (FC) theory and operating conditions. The advantages of increasing FC operating temperature and pressure are very important in this investigation. Literature discussing experiments in the variation of these parameters is included. In addition, literature was reviewed on similar work being performed using turbocharger type systems to recover the energy in FC exhaust. Also literature on other turbocharger type systems used for moisture and energy recovery was reviewed. The most promising was literature on all-air cooling and dehumidification systems for aircraft and automobiles. These systems are very similar to the system used in this experiment. This literature is summarized in addition to FC specific turbocharger literature. Also in the experiment, information about gas-liquid separators was needed. A summary of literature on separator theory and performance characteristics is also included.

## IV.1 Fuel Cell Moisture and Energy Recovery Literature Review

The following section will provide summaries of the literature reviewed on FCs and methods of moisture and energy recovery from FC exhaust streams. This literature was reviewed to determine typical PEM FC operating conditions and the effect of operation at higher pressures and temperatures. Included in some of the works is information on FC exhaust moisture and energy recovery systems presently in place or being experimentally investigated. Also include in this section are some examples of turbocharger type systems already developed.

Hirshenhoffer, *et al.* (1998) offers an overview of general FC theory and operation. This paper contains information on various types of FC and what distinguishes each type. This literature provides information regarding the typical operating conditions for all FC types and explains the basic theory behind each. Theory specific to the Proton Exchange Membrane (PEM) type FC was also gathered from this literature. Also discussed is the increased FC

performance associated with running at higher temperatures and pressures and the FC economic and structural impacts involved.

Brown and Jones (1999) prepared a feasibility study of fuel cells for stationary power applications. This study identifies typical operating conditions for each FC type and discusses the advantages and disadvantages of each for stationary power applications. The information on typical operating conditions was within the ranges suggested by Hirchenhoffer (1998) and the same characteristics of PEM FC were discussed. It was determined from this study that PEM FCs were very attractive for stationary power applications due to the safety of operating at a low temperature and having a solid, non-corrosive electrolyte.

Amphlett, *et al.* (1992) conducted experiments on the affect of operating conditions on PEM FC performance. The investigation was performed using a Ballard Power Systems Mark IV PEM FC. Tests were performed on a single cell and a 54-cell stack. The parameters varied were temperature, pressure, excess fuel, and excess air. This experiment was performed by identifying baseline operating conditions and varying one variable while maintaining the rest at the baseline conditions. This allowed the specific affect of each parameter to be observed independently. The results showed a nearly linear increase in power output in both the single and 54-cell stack for pressure and temperature increase. The findings also showed almost no power output increase in either stack configuration for excess fuel or air. However, the necessity of excess air for proper water and gas balance was discussed.

Amphlett, *et al.* (1998) also performed a computer simulation of a 250 kW PEM fuel cell system as a follow-up to the parametric study performed in 1992. The issues discussed include the need for moisture and energy recovery from the coolant exhaust stream as well as the actual FC exhaust. A computer simulation of the entire FC system, including the fuel reformer was performed and optimized. The findings show, an increase in overall system efficiency for increasing pressure and temperature. The importance of recovering energy and moisture from the exhaust streams to improve overall system efficiency was also discussed. The moisture recovery from the FC exhaust stream was assumed for the simulation to be performed prior to passing through the turbine with no method of performing this task discussed. A turbocharger

type system was modeled for energy recovery from the FC exhaust stream. The turbocharger type system was modeled to extract the energy from the exhaust stream and use this energy to compress the FC inlet air. The turbocharger system yielded good results in this model.

Pojoski (1999) discusses the design of a 250 kW power station utilizing a PEM FC as the sole power source. This test system is being built and operated in Berlin and will be used to supply power to the residential power grid if successful. The design of the system is discussed with emphasis on moisture and energy recovery from the exhaust streams. The system utilizes a heat exchanger to recover the energy from the FC coolant water stream exit. The heat exchanger uses the waste heat energy from the cooling system stream to preheat water entering the district hot water boiler. The FC exhaust is sent to the reformer where the moisture content is used to humidify the hydrogen fuel entering the FC. The high-temperature, high-pressure air is then sent to the turbine side of a turbocharger. The shaft work from the turbine is used to power the compressor side of the turbocharger. The compressor is used to supply the FC inlet air. This system is similar to the system used in this experiment.

Hagan, *et al.* (2000) discusses PEM FCs for automotive applications. The topics of water recovery and heat rejection are investigated. The system described in this paper utilizes a heat exchanger to drop the temperature of the FC exhaust stream below the dew point and condense out water to be reused by the FC. Based on FC operating conditions, the temperature the exhaust stream must reach to assure enough water condenses out to supply the FC moisture requirements is calculated. The findings show that operating the FC at pressures near atmospheric requires an unreasonably large heat exchanger to cool the exhaust enough to condense out sufficient water for water balance. The benefits of high-pressure operation are discussed. Raising the pressure of the exhaust stream has the effect of increasing the dew point temperature of the stream and thus increasing the temperature required for water balance. This would allow a reasonable size heat exchanger to be used in automotive applications. The amount of heat rejection is also discussed and is within the range discussed by Amphlett (1992) and Pojoski (1999).

Pentestar Electronic (1995) performed a study on optimal PEM FC design for automotive applications. The optimizing parameters are excess fuel, excess air, operating temperature, and

operating pressure. Excess fuel and air are identified as having no effect on FC performance but as being essential for gas and water balance. It is determined that the optimal pressure is 0.2 MPa (30 psi) at 80 C (ca. 180 F) for the PEM type FC in the size range suitable for automotive applications. A turbocharger was selected as the optimal choice for the compression of the air stream entering the FC. The FC exhaust stream, much like the design in this experiment, powers the turbocharger in this study. Much discussion is included in the report about the gains in system efficiency if the optimum turbocompressor could be developed.

McTaggart *et al* (1998) designed and tested a turbocharger type system for supplying compressed air to the FC inlet. The system uses the high-pressure FC exhaust to drive a turbine that in turn drives a compressor. The compressor compresses the air for the FC combustion process. It was found, that the amount of compressed air required could not be produced using the energy recovered from the exhaust stream alone. In order to achieve the required flow rate of air the compressor must be supplemented by a DC motor. The system is the same as the system developed in this experiment with the exception of the external motor and the fact that water recovery is not considered. The findings showed the need for a turbocharger system using no contaminating lubricant.

Honeywell (1998) developed a turbocharger type system for use with FC technology. This system works on the same principle as the Arthur D Little system The only difference is the development of an air bearing which eliminates possible air contamination from the turbocharger.

This section has offered information on FC operating conditions and moisture and energy recovery systems in use presently on FC exhaust streams. The agreement within the literature is very good and provides a reliable background upon which to perform FC research. The literature on turbocharger type systems offers many different schemes in which a turbocharger is the main component. This literature provides many ideas and insights into possible turbocharger type system configurations as well as the reasoning behind each.

### IV.2 Air Cycle Refrigeration Systems

This section will summarize pertinent information from the literature reviewed in the area of air cycle refrigeration systems. Air cycle systems are primarily in use on military and commercial aircraft. All-air refrigeration systems use no refrigerants and produce the desired cooling effect by removing work from a pressurized air stream. The main component of an air cycle refrigeration system is a turbocharger. The design of all air systems is very similar to the system developed in this investigation.

ASHRAE (1999) discusses all-air systems used to air-condition aircraft. The most common types of air-conditioning cycles in use on commercial and military aircraft are listed. The types include; bootstrap cycle, simple bootstrap cycle, chilled re-circulation cycle, and condensing cycle. The topic of moisture separation is also discussed in detail. In a typical system, engine bleed off air is entrained in the compressor side of a turbocharger. The compressor pressurizes the air and forces it through an air-cooled heat exchanger. The temperature reduction due to the heat exchanger causes moisture to condense out of the air stream. A high-pressure moisture separator removes the condensed water droplets from the air stream. The air then passes through the turbine side of the turbocharger and is further cooled. A low-pressure membrane type moisture separator is then employed to remove the water droplets condensing in this second cooling stage of the cycle. This air is then mixed with re-circulated cabin air, and introduced into the aircraft.

Grabow *et al* (1986) describes the use of a ram driven air cycle cooling system for aircraft cockpit and electronics. The cycle described is a reverse bootstrap cycle. In this system, air from outside the fast moving aircraft (ram air) is forced into the aircraft and through a moisture removal device. After the moisture is removed, the air stream is passed through the turbine side of the turbocharger system. The removal of energy from the ram air stream by the turbocharger cools the air. The cooled turbine exhaust air is used to remove waste heat from the cockpit and electronics. The air is then removed from these compartments by the compressor. Compressor exhaust is discharged out of the aircraft completing the cycle. One other major issue discussed in this paper is the characteristics of the turbocharger used in this system. The

turbocharger selected for this application has a sealed air bearing to prevent oil contamination of the cooling air. Also it is not feasible to remove all of the moisture content of the ram air; therefore, the turbine side of the turbocharger must be durable enough to handle condensing water. The ability to handle water condensation was one of the main considerations in turbocharger selection for the FC exhaust application described in this report.

Bhatti (1998) performed research on open-air cycle air conditioning systems for motor vehicles. The open-air cycle system is compared with the conventional R-134a system used in automobiles in terms of performance and environmental impact. The proposed open-air cycle system design is described in detail. The system consists of a turbocharger type system in which process air is entrained by the compressor. This air is then passed through an air-cooled heat exchanger to pre-cool the air before entering the turbine. Following the heat exchanger, the air enters the turbine where energy is removed from the air stream in the form of work further reducing the air temperature. The turbine shaft work is used to drive the aforementioned compressor. A centrifugal type separator performs the dehumidification of the air before it enters the passenger compartment. The system described in this paper is similar to the system used in the experiment performed in this report. However, the use of a heat exchanger before the turbine is not necessary in this experiment. The heat exchanger would reduce the effectiveness of the system described in this report due to the reduction of possible energy recovery by the turbine.

This section has provided general air cycle refrigeration system theory and characteristics. The similarities and differences between the system developed in this experiment have been highlighted as well. The literature reviewed in this section provides a good background for developing an all air cycle for fuel cell moisture and energy recovery. The turbocharger system described in this experiment is merely a modification of the systems described in this section.

**IV.3 Separation Literature Review**

This section provides a summary of literature reviewed for the moisture separation phase of the experiment. In the system developed in this investigation the exhaust from the turbine

side of the turbocharger is a gas-liquid mixture. A centrifugal type separation system was developed in order to remove water droplets from an air stream. The following literature provides a foundation for understanding separation techniques with an emphasis on centrifugal separation. Issues of importance include pressure drop through the separator and droplet capture efficiency.

Leith and Licht (1972) provide a mathematical approach to predicting centrifugal separator performance. The study is performed with special emphasis on the phenomena of back mixing. In this approach, particles that come into contact with the wall or settle into the bottom of the separator are not assumed captured and are still considered in the analysis. This article describes in detail, particle motion, distribution, and collection within a centrifugal separator. The equations of cyclone performance are compared with published data and yield favorable comparison. The collection efficiency and pressure drop associated with a cyclone separator is also discussed. The merits of optimization of collection efficiency with minimal pressure drop are explored. A method using the equations developed to take both parameters into account when designing a cyclone separator is given.

Setford (1995) provides an overview of separation techniques and applications. Included, are separation techniques for applications commonly encountered in the engineering discipline. The issue, of removing liquids from gas streams, is discussed in detail and many possible separation techniques to perform this task are described. The techniques described include impingement, membrane, and centrifugal separation. The centrifugal separator is determined to be very efficient for gas-liquid applications. The justification given for the high efficiency liquid removal is liquids coalesce on the cyclone wall when they strike. This reduces the possibility of re-entrainment of the substance in the exhaust stream, which is a major problem in solids removal. Also discussed is the relatively low maintenance and pressure drop associated with centrifugal type separators.

Griffiths and Boysan (1996) performed experiments combining empirical and computational models of various cyclone separators. Collection efficiency and pressure drop are discussed in great detail and are identified as the measures used by the authors to determine

separator performance. In this experiment computational techniques are used to predict separator performance and are compared to empirical data. The computational predictions were very close to the empirical results. This method is recommended for use in future separator design.

Avci and Karagoz (2000) developed a mathematical model to predict cyclone separator performance. The effects of cyclone geometry, surface roughness and concentration of particles were the main parameters considered. Separator pressure drop and capture efficiency were the main performance characteristics upon which the optimization was performed. The model predicted the performance well based on other similar studies and experimental results. The findings show, cyclone height and fluid inlet velocity are the primary factors in centrifugal separator performance.

This section presents literature providing general theory and operating principles of separators with an emphasis on the centrifugal type. Literature reviewed on separation seems to be in very good agreement on the topic of performance measures. The primary performance characteristics in separation selection are capture efficiency and pressure drop. From the literature, it is also evident that the lowest pressure drop and highest capture efficiency for gas-liquid systems is achieved by the centrifugal type separator and that is why it was chosen for this application.

# CHAPTER V

## APPARATUS AND INSTRUMENTATION

The following section describes the apparatus used in this investigation. The objective of the research was to design and test a system to recover the moisture and energy in a Proton Exchange Membrane (PEM) Fuel Cell (FC) exhaust using a turbocharger and moisture separator. In order to model the exhaust stream, a test facility was developed to provide the required flow rate of compressed air and a resistance heater to heat the air. Moisture is then added to the air as saturated steam before it enters the prototype moisture and energy recovery system. This prototype recovery system consists of an automotive type turbocharger and a moisture separator. The turbocharger must be constantly supplied with oil for lubrication and heat removal. For this reason, an oil delivery system was also designed and built for this apparatus. This oil pumping, piping, and heating system ensure the oil is delivered to the turbocharger at the correct temperature, pressure, and flow rate. Another key component of the testing apparatus is the instrumentation. The instrumentation system allows for the measurement of all needed parameters in order to fully characterize the system performance. Thermocouples, a relative humidity (RH) probe, pressure gauges, and rotameters were used to collect data in this experiment. The readings taken in this experiment are outputted to a personal computer based data collection system. In summary, the test apparatus consists of the four subsystems listed below.

1. Moist air stream modeling system (i.e. heater, piping, and boiler)
2. Moisture and energy recovery system (i.e. turbocharger and separators)
3. Oil supply system for the turbocharger (i.e. pump and piping)
4. Instrumentation and data acquisition system (i.e. thermocouples, RH probe, pressure gauges, and rotameters)

A schematic of the test rig without the oil system is displayed in Figure 5-1, and the components are listed along with the manufacturer and model number below in Table 5-1.

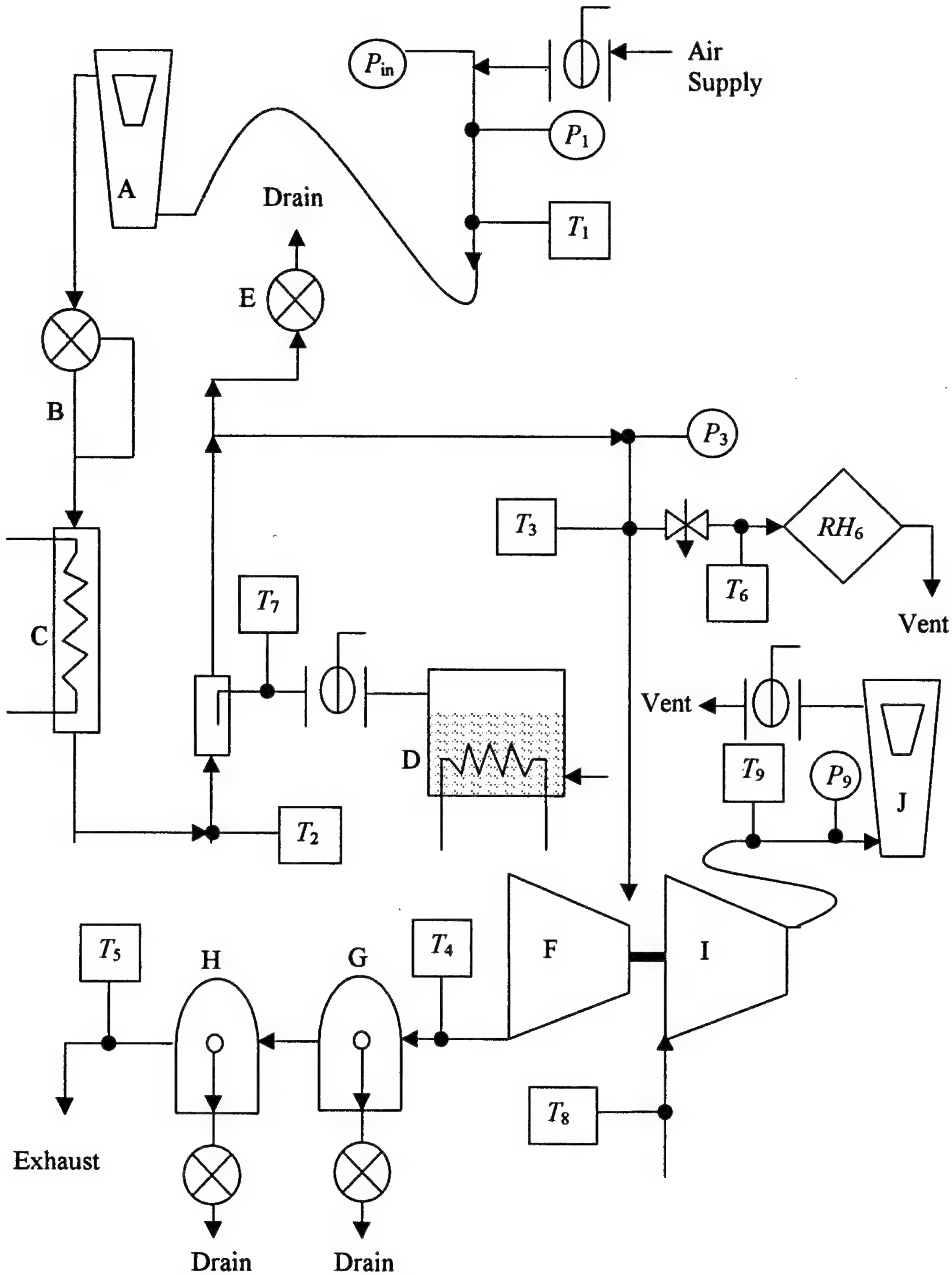


**Figure 5-1 Schematic of test rig.**

**Table 5-1 Apparatus designations and descriptions.**

| Block | Description | Manufacturer | Model Number |
|---|---|---|---|
| A | Rotameter | Schutte-Khoerting | 6-HCFB |
| B | Pressure Regulator | McMaster-Carr | R17-801-RGLA |
| C | Air Heater | Chromalox | FTS-048475 |
| D | 20 KW Boiler | Electro-Steam | LB-20 |
| E | Steam Trap | Spirax-Sarco | B-1H |
| F | Turbine | Garrett | GT-1541V |
| G | Separator 1 | Hayward | Type T |
| H | Separator 2 | Hayward | Type T |
| I | Air Compressor | Garrett | GT-15 |
| J | Rotameter | Fischer-Porter | 10A3565A |

**VI.1 Moist Air Stream Modeling System**

This section will describe in detail the portion of the test apparatus designed to model the exhaust stream of a PEM FC. The purpose of the moist air stream modeling portion of the testing apparatus is to produce an air stream in which temperature, pressure, humidity, and flow rate can be controlled in the ranges needed for this experiment. Control of each parameter is desired so that a parametric study can be performed on the prototype. A drawing of the moist air stream modeling portion of the test rig using apparatus designations from Table 5-1 is shown below in Figure 5-2.

The apparatus is mounted on a steel channel frame (Unistrut Metal Framing System) chosen for its structural rigidity and expandability. The piping on the inlet side of the turbocharger is all brass. Brass piping was chosen primarily for its corrosion resistance, as no debris is tolerable through the turbine. Brass piping also has adequate structural properties. Strength of piping material is important, as in many places the piping is the load-bearing member of the structure.

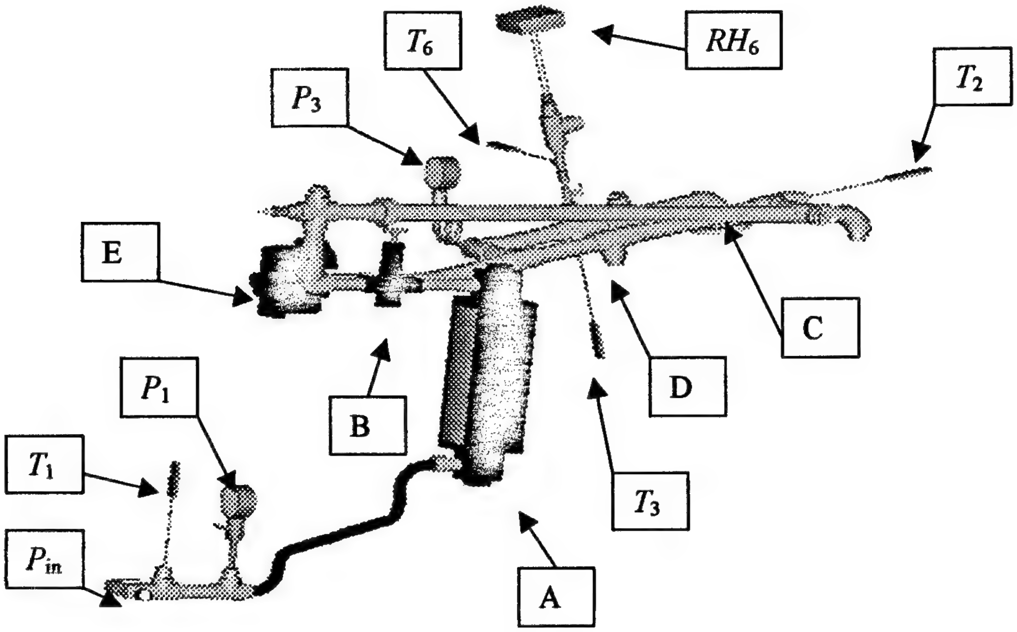


**Figure 5-2 Moist air stream modeling system.**

The building air supply is supplied to the lab in a 19 mm (0.75 inch) outer diameter (OD) pipe. It is then metered into the system through a ¾ inch nominal pipe size, NPT threaded, brass ball valve with 26.67 mm (1.050 inch) OD and 20.93 mm (0.824 inch) Inner Diameter (ID). From the ball valve, the inlet air flows into a female tee. This tee allows the inlet air pressure ($P_{in}$) to be measured while providing a means for the air to continue into the rest of the test rig. Following the tee, the piping size is increased to 1 inch nominal pipe size, NPT threaded, with 33 mm (1.315 inch) OD and 27 mm (1.049 inch) ID. The airflow continues into the system through two brass tees connected by a nipple. The tees allow a thermocouple ($T_1$) and a pressure gauge ($P_1$) to be mounted into the air stream. Flexible 1 inch OD hose fitted with 1 inch nominal pipe size, NPT threaded, brass end connectors, attaches this wall-mounted section to the rotameter on the main section of the rig. Flexible hose was chosen for this application, to allow the test rig some level of maneuverability. The flexible hose also provides a convenient means of disconnecting the main rig structure from the wall if for any reason it needs to be temporarily relocated.

The flow meter is a variable area type rotameter (Schutte Khoerting model 6-HCFB). The inlet and outlet connections are 1 inch nominal pipe size, NPT threaded. The purpose of the rotameter is to gauge the mass flow rate of the test apparatus air, in units of SCFM. The rotameter has been modified to measure the test apparatus flow, by replacing the tube (McCrometer model 6-HCFB) and float (McCrometer model 64-J) to increase the flow measurement capabilities. The rotameter is now capable of measuring airflow up to 110 SCFM at 0.322 MPa (32 psig) and 21.1 C (70 F). To support the rotameter, the nipple at its outlet is rigidly supported by a steel channel beam (Unistrut Metal Framing System). The nipple also serves to support the pressure regulator, which is not suitable for load bearing.

The pressure regulator is a 1 inch nominal pipe size, NPT threaded, compressed air regulator (Mc Master-Carr model 4959K5) with a pressure range of 0.135 to 1.14 MPa (5 to 150 psig). The pressure regulator allows for control of the air pressure into the apparatus and also provides a means for the turbine inlet pressure to be controlled and kept constant. Another nipple immediately follows the pressure regulator and allows the air to flow into a reducing tee where the pipe size increases to 1 ¼ inch nominal pipe size, NPT threaded, with 42.12 mm (1.66 inch) OD and 35.05 mm (1.38 inch) ID. Another pipe nipple connects the reducing tee to a cross.

The cross serves as an enclosure for the cool end of the heater. This heater end is secured here with a bored through ½ inch nominal pipe size, NPT threaded, Swagelock 12.7 mm (.5 inch.) tube fitting. The air heater is a finned tube resistance-heating element (Chromalox FTS-048475) mounted inside the pipe. A PID temperature controller (Omega model CN9000A) energizes and de-energizes the heater to keep a preset air temperature. The temperature measurement used by the controller is a thermocouple ($T_3$) mounted just upstream of the turbine. This allows the turbine inlet temperature to be held constant and be easily variable. The remainder of the heater is enclosed inside two sections of pipe. A coupling, for ease of dismantling, connects the pipe lengths.

The hot end of the heater is free to move within the pipe along the heater length to provide for heater thermal expansion. The heater is restrained from moving perpendicular to its length by a rigidly supported pipe hanger. This pipe hanger constrains the heater end in this

direction. The electric wires are attached to this end by inserting them into a short piece of polymer type (Tygon) tubing. The Tygon tubing is sealed into the piping with a ¼ inch nominal pipe size, NPT threaded, Swagelock 6.35 mm (0.25 inch) tube fitting. The wire lead is then attached to the end of the heater inside the pipe.

From the cross housing the hot end of the heater, the airflow is forced into two 90 degree turns making the test apparatus air flow in the opposite direction as before the cross. A tee connected to the cross by a nipple accomplishes this. The nipple is rigidly connected to the steel channel frame (Unistrut Metal Framing System) by a pipe mount. The tee connected to the other side of the nipple allows for a thermocouple ($T_2$) mount.

Following the bend, the air flows into two pipes connected by a tee where the packaged boiler (Electro-Steam model LB-20 serial no. 31937) is connected to the system. The purpose of the boiler is to feed saturated water vapor into the heated test apparatus air stream to humidify the air. The steam flows out of the boiler through a ball valve, and into ½ inch OD copper tubing. A "U" bend in the tubing allows for thermal expansion. The tubing is mounted into the pipe with a bore through ½ inch nominal pipe size, NPT threaded, 12.7 mm (0.5 inch) Swagelock tube fitting. A 90 degree bend in the tubing immediately after entering the pipe allows for injection of the steam against the airflow forcing proper mixing.

Following the boiler inlet, a tee offers a drain for any condensed steam. The piping size is reduced to ½ inch nominal pipe size, NPT threaded, pipe and connects to an inverted cast iron bucket steam trap (Spirax Sarco model B-1H). The piping is arranged so the steam trap openings are horizontal, as orientation of the trap is essential for its operation. The steam trap provides a means of pressurizing the system and still collecting water. The steam trap is checked periodically to see if water is condensing.

The air stream takes another bend through the other outlet of the drainage tee and into a pipe nipple mounted rigidly to the steel channel ram with a rigid pipe clamp. The stream continues into another tee where a pressure gauge ($P_3$) is mounted. The other branch of the tee is connected to a pipe nipple held in place by a pipe hanger to give support for the turbine. This

nipple also connects to a cross allowing for insertion of a thermocouple ($T_3$) and allows for bleed off of air for the Relative Humidity ($RH_6$) probe.

The RH probe must not be exposed to pressures exceeding 0.101 MPa (0 psig). To accommodate this pressure requirement a small amount of the turbine inlet air is bled off and throttled down to atmospheric pressure with a needle valve. This air is then piped through two ¼ inch nominal pipe size, NPT threaded, hex nipples connected by a tee. This tee allows a thermocouple ($T_6$) to be mounted in the bleed off air stream. The pipe size is then increased to 1 inch nominal pipe size and flows through a tee, where the RH probe is mounted, and the bleed off air is allowed to escape.

The air not bled off, flows through a pipe nipple and into a 90 degree elbow allowing the turbine to be mounted in the correct orientation. A pipe nipple connects the elbow to a flange onto which the turbocharger is bolted. It is important to note that the turbocharger is designed to only be supported by this inlet flange. At this point the air stream enters the turbocharger and thus the prototype portion of the apparatus.

### V.2 Turbocharger and Centrifugal Separator

This section will provide detail on the mounting, piping, and instrumentation associated with the turbocharger and separator system used in this experiment. A drawing of this portion of the system, using apparatus designations from Table 5-1, is shown in Figure 5-3.

The high humidity air stream from the FC exhaust modeling portion of the testing apparatus is passed through the turbine side of the turbocharger (Garrett model GT1541V). The turbine removes energy from the air stream and reduces the pressure and temperature of the mixture. The loss in pressure and temperature of the fluid is a desirable effect as it drops the mixture below the dew point and causes the water in the stream to condense. The energy the mixture loses through the turbine is converted to shaft work. The shaft work from the turbine is used to compress ambient room air with the compressor side of the turbine.

The inlet of the compressor side of the turbocharger operates in vacuum pulling air in from the lab. The compressor must be protected from foreign objects being entrained in the inlet flow. To ensure no potentially harmful objects get into the compressor, an air filter (Mc-Master-Carr model 4399K45) is in place at the inlet. The filter has a 1 ½ inch nominal pipe size, NPT threaded, connection at the exit. The filter is connected to the compressor inlet by a PVC nipple. The PVC nipple is connected to the compressor inlet by a flexible pipe coupling. A thermocouple ($T_8$) is mounted through the wall of the PVC nipple to attain the inlet air temperature. The room air passes through the compressor, into clear flexible reinforced hose, and into stainless steel 1 inch nominal pipe size, NPT threaded, piping. Two tees are connected by a nipple and allow for the insertion of a thermocouple ($T_9$) and a pressure gauge ($P_9$). The air stream then passes through a rotameter (Fischer-Porter model 10A3565A serial 6801F1002B1) measuring the air flow through the compressor in units of SCFM. Immediately after the rotameter, a ball valve is mounted to allow the air stream to be throttled. The throttling valve allows the compressor to be loaded to maximize the amount of work done by the turbine

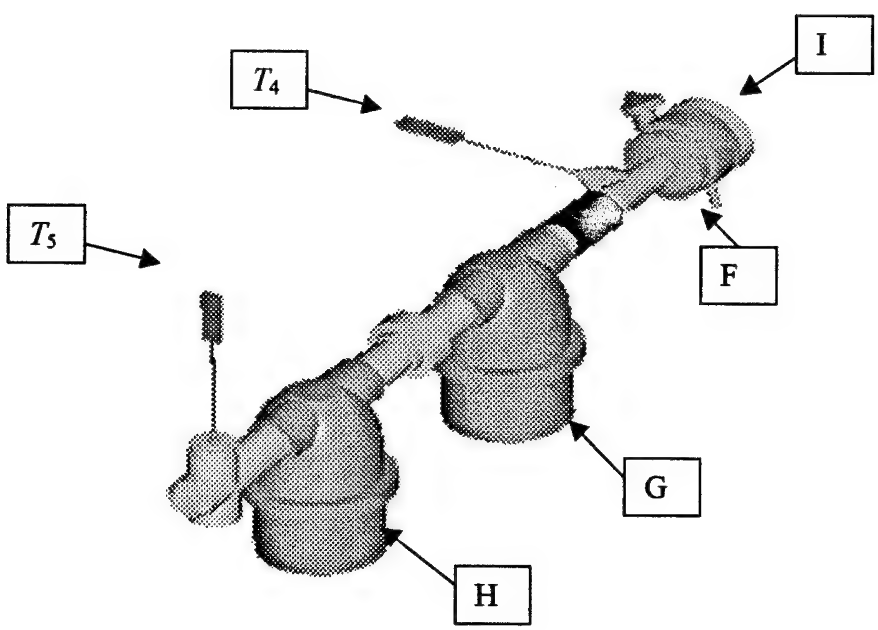


**Figure 5-3 Assembly drawing of prototype system.**

The air that passes through the turbine from the moist air modeling system enters into two 1 ¼ inch nominal pipe size, NPT threaded, nipples connected by a tee. This tee allows for the insertion of a thermocouple ($T_4$) immediately after the turbine. The nipple furthest from the turbine is clamped to a flexible reducing pipe coupling which makes the transition from 1 ¼ to 2 inch nominal pipe size. A flexible coupling was chosen since the turbocharger is designed to only be constrained by the turbine inlet mount.

The air then enters the liquid gas separator(s) (Hayward model Type T) where the liquid is removed and caught in a steam trap (Hayward model 90AC) mounted directly below the separator. The steam trap is designed to open when the water level inside reaches a certain height. Once the air passes through the separator(s) it enters into a cross where a thermocouple ($T_5$) is mounted. The air is then piped out of the lab window through clear reinforced PVC hose. To prevent uncollected water droplets from being propelled into the walkway beneath the window a "U" bend is created at the end of the hose. The bend is accomplished by inserting a bent section of copper tube into the end of the PVC hose. This bend forces the moist exhaust stream exiting the hose to immediately strike the wall of the building outside. This eliminates the possibility of moisture being sprayed onto the walkway below.

### V.3 Turbocharger Oil Supply System

This section will describe the general system designed to supply and remove oil from the turbocharger. Figure 5-4 is a drawing of the oil supply system, with apparatus designations from Table 5-1.

The turbocharger is designed for use in an automobile engine. In order to properly lubricate the turbocharger bearings, oil from an automobile oiling system must be simulated. The turbocharger manufacturer recommends SAE 10W-30 motor oil be supplied to the turbocharger at 0.3 gallons per minute, 0.2 to 0.5 MPa (30 to 70 psig) pressure, and 40 to 90 C (*ca.*100 to 200 F). One thing important to note about the turbine is the oil must exit at atmospheric pressure. Therefore, the outlet drain must be configured so there is no restriction at any point in the line that would create backpressure.

A 610 mm (24 inch) long, 2 inch nominal pipe size, NPT threaded, galvanized steel, pipe nipple was used for an oil reservoir. This allows for easy attachment of the other necessary piping for oil circulation. The upper portion of the reservoir is attached to a 2 inch nominal pipe size tee. The tee allows for the addition of a nipple, attached in-line with the reservoir nipple. The pipe size is then reduced to ½ inch nominal pipe size by a bell reducer through which the oil bypass empties. The bell reducer opening fits snuggly around the oil bypass line preventing oil contamination from airborne debris. The tee opening perpendicular to these two pipes is used for mounting a horizontal nipple. This nipple serves as the drain from the turbine oil outlet back to the oil reservoir. This nipple connects to a 90 degree elbow with a 2 inch long nipple at the opposite end. This nipple serves to prevent oil splash from the gravity fed turbine oil exhaust

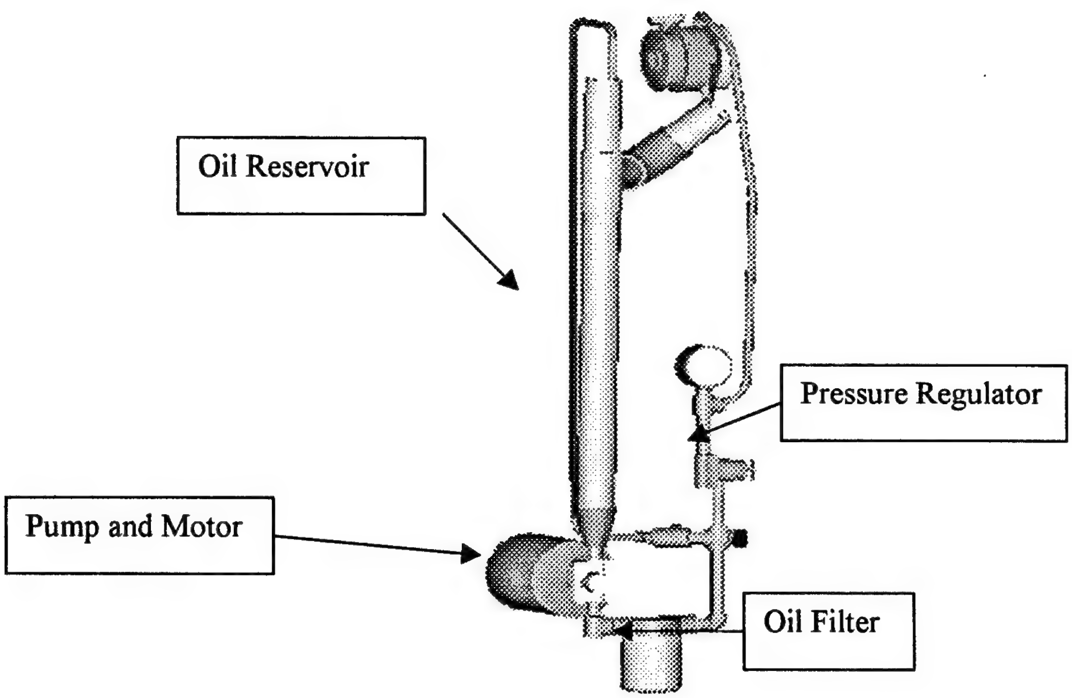


**Figure 5-4 Oil supply system.**

The bottom of the oil reservoir is where the oil exits, and enters the pump. The pipe size is decreased, from 2 inch nominal pipe size to ½ in nominal pipe size, to accommodate the pump inlet. The pump chosen is a fat fryer gear pump (Teel model 2P389D Serial no. 1093). This pump is designed to handle oils at elevated temperature, as was encountered in this application. The pump is connected to the reservoir by a nipple. The pump outlet goes immediately into a tee where a drain plug is installed to allow for system draining. The piping from the other tee opening is increased to ¾ inch nominal pipe size where it enters the oil filter (Hydac model 0080MA010BN). The filter removes particles greater than 15 microns in size as was recommended by the turbocharger manufacturer. Piping size on the other side of the filter is reduced again down to ½ in nominal size. The oil is then enters a 90 degree elbow where the oil begins to again flow vertically. The oil is then passed into a cross. To measure the oil temperature a gauge thermometer is mounted in one cross opening perpendicular to the flow.

The cross opening inline with the flow allows for some of the oil to pass through and eventually reach the turbine. However the pumping capacity of the pump is much too great for the turbine and some oil flow must be bypassed. The other cross opening allows oil to be returned directly to the reservoir. The oil flow is metered by a ball valve, and routed back to the oil reservoir, through ½ inch OD copper tubing. Constant circulation of the oil serves to heat it to within the acceptable turbine inlet temperature range specified by the turbine manufacturer. An additional heating system is utilized to attain higher oil temperatures. Circulating the oil at elevated temperature is currently being tested to determine if this prevents or eliminates water in the oil supply. Oil contamination will be discussed later in this report. The heating system consists of flexible pipe wrap heating tape secured around the oil reservoir. The temperature of the circulating oil is controlled by a variac (Powerstat model 116B) accepting 110V input.

The oil that does not bypass the turbocharger passes through a pressure regulator. The pressure regulator (Watts model 2A65) controls the pressure of the oil flow into the turbocharger. This turbocharger oil inlet pressure is monitored by a pressure gauge mounted just downstream of the regulator. As stated previously, the turbocharger can only be restricted by the turbine inlet flange. To ensure no part of the oiling system constricts the turbine, a flexible oil line is used to transport the oil from the piping system to the turbocharger oil inlet.

## V.4 Instrumentation

### V.4.a Data Measuring Devices

This section will describe in-depth the instrumentation used in this experiment. The instrumentation in this experiment consists of 9 thermocouples, 3 pressure gauges, 2 rotameters, and a humidity probe. The relative humidity (RH) measurement was taken using a capacitance type probe (Vaisalla model HMD-60U serial no. W1040057) furnished with a certificate of calibration and calibration data. The probe is mounted into the air stream using a bored through ½ inch, NPT threaded, nominal pipe size, stainless steel, 12.7 mm (.5 inch) Swagelock tube fitting. Pressure gauges $P_1$ and $P_3$ are (Omega model PGT-30L-60) test gauges with a range of 0.101 to 0.515 Mpa (0 to 60 psig). Pressure gauge $P_9$ (Omega model PGT-30L-30) is the same but has a range from 0.101 to 0.308 Mpa (0 to 30) psig.

Seven of the thermocouples are ungrounded copper constantan, type T quick disconnect, with 3.175 mm (0.125 inch) 304 stainless steel sheath single element (Omega model CPSS-18U-12) thermocouples, and 2 are dual element of the same type (Omega model CPSS-18U-12-Dual). All 9 thermocouples are mounted in the air stream using 1/4 inch nominal pipe size, NPT threaded, bored through swagelock 1/8 inch tube fittings to eliminate leakage.

The thermocouples were calibrated using a constant temperature bath (Techne Inc., Model DB-35L, serial no. 0050774) to expose the thermocouples to and hold them at a constant temperature. The exact temperature of the bath is then determined with an RTD probe outputting to a digital multimeter (Hewlett Packard, model 34401A, serial no. US36064119). The temperatures read by the thermocouples are plotted against the exact temperatures to perform the regression yielding the calibration equation. The RTD probe temperature measurement is considered to be the exact measurement used for this regression. Linear and quadratic regressions were performed with the linear equation being slightly more accurate. Consequently, a linear regression model was used for the calibration equation. Equation 5-1 provides the calibration equation used to correct the thermocouple readings taken in this experiment.

$$T_{corr} = c + b\,T_{meas} \qquad (5\text{-}1)$$

| | | |
|---|---|---|
| Where: | $T_{meas}$ | temperature measured by thermocouples |
| | $T_{meas}$ | corrected temperature |
| | $c$ | intercept of thermocouple regression model |
| | $b$ | slope of thermocouple regression model |

The uncertainty associated with the thermocouple calibration equation is calculated by the spreadsheet used for the regression in terms of standard error ($u$). In order to get corrected temperatures in the 95% error band the expanded uncertainty ($U$) given in equation 4-2 will be used.

$$U = k_c u \qquad (5\text{-}2)$$

| | | |
|---|---|---|
| Where | $U$ | expanded uncertainty [C] |
| | $u$ | standard error [C] |
| | $k_c$ | coverage factor |

In this case the number of data points sampled is twelve resulting in ten degrees of freedom and consequently a $k_c$ value of 2.2. Another statistical method for determining the quality of the calibration equation is the alpha risk. The alpha risk is the chance that the regression model is not statistically significant. The correlation coefficient ($R^2$) is a measure of how closely the regression model fits the actual data and is included in this analysis. The calibration curves developed all are within reasonable standards and therefore our corrected thermocouple temperatures should be considered statistically reliable. The slopes and intercepts associated with the linear calibration equations for each of the nine thermocouples used in this experiment along with the expanded uncertainty, alpha risk, and correlation coefficient associated with each is displayed in Table 5-2.

**Table 5-2 Thermocouple calibration data.**

| Thermocouple | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|---|
| Intercept (C) | 1.437 | 1.240 | 0.774 | 0.802 | 0.257 | 0.083 | 0.526 | 0.100 | 0.009 |
| Slope | 0.961 | 0.965 | 0.982 | 0.966 | 0.981 | 0.9828 | 1.009 | 0.980 | 0.982 |
| $U$(C) | 0.341 | 0.265 | 0.101 | 0.304 | 0.098 | 0.0865 | .0447 | .2478 | .1138 |
| $R^2$ | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 |
| Alpha Risk × $10^{27}$ | 1169 | 94.10 | 375.7 | .0051 | .0036 | .0010 | 64.31 | 17.62 | .0238 |

All of the relevant instrumentation is listed in Table 5-3 below, along with manufacturer information.

**Table 5-3 Instrumentation information**

| Measurement | Description | Manufacturer | Model number |
|---|---|---|---|
| Flow Rate (A) | Rotameter | Schutte-Khoerting | N/A |
| Flow Rate (K) | Rotameter | Fischer-Porter | 10A3565A |
| Temperature | Thermocouple | Omega | CPSS-18U-12 |
| Temperature | Thermocouple | Omega | CPSS-18U-12-DUAL |
| Pressure ($P_1$,$P_3$) | Pressure Gauge | Omega | PGT-30L-60 |
| Pressure ($P_9$) | Pressure Gauge | Omega | PGT-30L-30 |
| Relative | RH Probe | Vaisalla | HMD-60U |

The first measurement along the air stream is the pressure gauge $P_{in}$, mounted at the compressed air inlet of the system. This gauge is very inaccurate, and serves only to give a rough estimate of the line pressure entering the system. The second gauge $P_1$ is a more exact test gauge having a maximum pressure capacity of 0.52 MPa (60 psig). This gauge is sized to measure the steady state pressure of the line. $P_1$ is protected by a ball valve, remaining closed until $P_{in}$ reaches a pressure within its range. This allows for the inlet pressure to be measured accurately without damaging $P_1$.

The pressure gauge $P_1$ and thermocouple $T_1$ are in place to monitor the rotameter inlet conditions. The rotameter is calibrated for 0.33 MPa (32 psig) and 21.1 C (70 F). Any deviation from this inlet condition requires a correction calculation. $T_1$ and $P_1$ provide the only information needed for this calculation, allowing for the actual flow rate to be calculated.

Thermocouple $T_2$ is in place for a future application. Knowing the temperature on both sides of the heater and knowing the power supplied to the heater, allow the confirmation of the flow rate measurement from the rotameter. This system is not in place at this point but will be in the future. $T_2$ is a dual element thermocouple with only one output utilized. This output goes to the data collection system.

The turbine inlet conditions are measured using three instruments. Pressure gauge $P_3$ is mounted on a 270 degree pigtail allowing the pressure to be measured after the steam input without the gauge being exposes to the steam in the pipe. Thermocouple $T_3$ measures the temperature at the turbine inlet. This thermocouple is a dual-element type, with one output going to the data recorder and the other going to the heater temperature controller. The use of a dual-element thermocouple allows the temperature at the turbine inlet to be recorded and controlled, using only one thermocouple.

The RH probe ($RH_6$) is used to measure the turbine inlet humidity. This probe must not be exposed to pressure exceeding 0.101 MPa (0 psig). In order to take the RH measurement, the high-pressure turbine inlet air is bled off, and throttled through a needle valve down to atmospheric pressure. This air is then passed over thermocouple $T_6$ to monitor the temperature after the throttling process, ensuring the process is adiabatic. The air then passes over $RH_6$ where the humidity measurement is taken. To ensure air is passing over the probe at all times the bleed off stream empties in to a water-filled container. The water in container allows the exit of the exhaust gas to be observed as bubbles.

After the turbine, two thermocouples are utilized to attain the temperatures in the moisture removal portion of the test facility. Thermocouple $T_4$ monitors the temperature

immediately after the turbine expansion process. Thermocouple $T_5$ is placed immediately after the separator stage of the system.

Thermocouple $T_7$ is mounted in the boiler exit stream to monitor the steam temperature entering the test apparatus. This measurement is important for an energy balance on the system and to determine the state of the steam used to humidify the air. The boiler features a pressure gauge from the manufacturer that monitors the steam pressure.

The condition of the air before and after the compressor side of the turbocharger is monitored to determine pressure, temperature, and flow rate. Thermocouple $T_8$, is in place to measure the inlet temperature of the compressor gas. The inlet pressure is assumed to be the atmospheric pressure measured by a barometer (Swift Instruments model 477). The flow rate is measured in units of SCFM by a rotameter (Fischer-Porter model 10A3565A serial 6801F1002B1) mounted just downstream of the compressor. The rotameter is calibrated for 0.101 MPa (0 psig) and 25 C (77 F). If the inlet temperature or pressure of the rotameter is different from the calibration temperature, a correction factor is needed to determine the actual flow rate. Inlet pressure and temperature are measured for the rotameter correction by pressure gauge $P_9$ and thermocouple $T_9$.

V.5.b Data collection system

This section will describe the data collection system and procedure used in this experiment. The data acquisition system (Agilent model 34970A) used in this experiment outputs to a Personal Computer (PC). The thermocouples output a voltage signal, and therefore can be connected directly to the data collector with no external circuitry. Using the inputted thermocouple type the data collector converts the thermocouple voltage reading into a temperature reading. The calibration offset and intercept can also be programmed into the data collection system so that calibrated data can be collected with this setup.

The RH probe is a capacitance type probe and must be connected to the data collector in the manner shown in Figure 5-5 below.

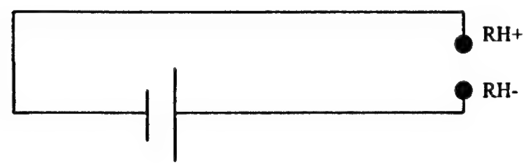


**Figure 5-5 RH probe configuration.**

The voltage source must supply 10 to 30 VDC to the probe at all times to achieve the 4 to 20 mA output. This is accomplished by wiring a stand-alone DC power supply (Sorensen model QRS40-.75) in series with the data collector and probe. The current read by the data collector is converted to percent RH using the data collector scale and offset command.

The sampling rate of the instruments is set in the PC to be one scan every three seconds. The data collected is already calibrated using the gain and offset function in the data collector for each device. The data collector scans all channels on three second intervals and outputs the values to a PC, where all values can be viewed at once. The wiring of the instrumentation to the data collector allows the module to scan in the order shown in Table 5-4.

**Table 5-4 Measurement sequence.**

| Measurement | Instrument | Channel |
|---|---|---|
| 1 | Thermocouple 1 | 101 |
| 2 | Thermocouple 2 | 102 |
| 3 | Thermocouple 3 | 103 |
| 4 | Thermocouple 4 | 104 |
| 5 | Thermocouple 5 | 105 |
| 6 | Thermocouple 6 | 106 |
| 7 | Thermocouple 7 | 107 |
| 8 | Thermocouple 8 | 108 |
| 9 | Thermocouple 9 | 109 |
| 10 | Relative humidity probe | 121 |

The data collected throughout the testing sequence is saved in the PC. This data is exported as a text file onto a floppy disk. The text file is then imported into Microsoft Excel for windows. In Excel the data can be analyzed and processed. The thermocouple and relative humidity probe readings are averaged initially over a period of ten scans to ensure slight variations do not affect the data value used. This data is inspected to see when equilibrium was reached in the experiment. Once equilibrium is reached the averaged data points are again averaged and that value is the reading for that instrument used in the final results.

# CHAPTER VI

## EXPERIMENTAL PROCEDURE

This section will discuss the experimental procedure used in this investigation. A short discussion is first provided to convey the underlying logic behind these procedures. The test rig is designed to perform moisture and energy recovery experiments. In order to perform these experiments in a way that is scientifically interesting, the test rig is designed to allow key system parameters to be varied. The rig also allows these parameters, once reached, to be held steady throughout an experiment. The variable parameters are temperature, pressure, flow rate, and humidity. To ensure repeatability of results, a consistent series of steps is followed before initiating measurements.

Aside from repeatability, there are other issues to be considered in the development of the experimental procedure. The main issue in this experiment is to make sure that operation of the equipment is within safe ranges to ensure no harm is done to the experimental equipment. One such consideration is allowing the oil supplied to the turbocharger to reach at least 38 C (100 F) before any air is sent through it. This ensures that the oil supplied to the turbocharger has the proper viscosity to lubricate the bearings inside. In addition, the air is shut off before the oil pump is deactivated at the end of experiments ensuring the turbocharger is never in operation without adequate oil supply.

Another consideration is that the heater used in this experiment must never be operated without sufficient air flowing across it. Running the heater without air flow is potentially harmful as the heater may overheat and burn out. For this reason, the heater is always deactivated before the airflow is shut off. The air is allowed to flow until the air downstream from the heater reaches room temperature. This ensures the heater will be run safely and no damage due to overheating this component will occur.

The boiler in this experiment reaches a somewhat high temperature and pressure. When deactivated the boiler is still able to supply steam to the system for a significant amount of time. This creates a problem if the airflow is stopped. Steam enters the system and is not circulated out of the system. The steam condenses on the cold pipes and can settle in the areas where the electrical connections enter the system. The settling of the water creates an equipment hazard as well as a potential personal safety hazard. Therefore the steam cutoff valve must be closed prior to shutting off the airflow.

The data collection system used in this experiment provides a continuous output of the instrumentation readings. These values are closely monitored to determine when steady state is reached. The data collector is then initialized to begin recording data. Monitoring only at critical times limits the data collected to the time when the apparatus has reached steady state or very close to it. Steady state operation is reached at two times during a typical experiment, therefore, the data collector must twice be initiated during a testing sequence. The data collector is initialized when the test apparatus has reached dry air steady state and again after the moist air reaches steady state.

To avoid any premature equipment failures, personal injuries, and to ensure repeatable results the following order of events are to be followed for taking data in this experiment.

1. Take a 25 mL sample of the oil from the oil reservoir and identify it appropriately.
2. Energize the turbine oil supply system and monitor the oil temperature constantly.
3. Open the ball valve to the air supply once the turbine oil temperature reaches 38 C (100 F).
4. Activate the air heater and controller, setting the controller to maintain the air temperature at the desired value.
5. Allow the system to run until steady state temperature, pressure, relative humidity, and flow rate are reached.
6. Initialize the data collector to record all steady state temperature, pressure, relative humidity, and flow rate readings.
7. Energize the boiler water pump and allow the boiler to fill with water.

8. Energize the boiler.
9. Allow the system to run until steady state temperature, pressure, relative humidity, and flow rate are reached.
10. Capture the condensation from the separator steam trap for two minutes.
11. Initialize the data collector to record all steady state temperature, pressure, relative humidity, and flow rate readings.
12. De-energize the boiler.
13. De-energize the air heater, monitoring the upstream temperature constantly.
14. Close the boiler outlet valve.
15. Close the air supply valve once the air temperature upstream of the heater reaches room temperature.
16. De-energize the turbocharger oil supply system.
17. Take a 25 mL sample of the oil from the oil reservoir and identify it appropriately

The first and last step of the testing procedure is to take samples of the oil in the oil reservoir. This facilitates monitoring of the quality of the oil used to lubricate the turbocharger. It was observed that some leakage across the turbine oil seal occurs allowing the oil to be contaminated by the moisture content of the entering air stream. To determine how much water is in the oil stream, the sample weights and the volumes are recorded. The oil samples are then heated until the water content of the mixture is removed. The weight and volume of the samples are again recorded. The change in weight and volume allow for calculation and comparison of the volumetric and mass percents of water in the oil before and after each experiment. This data also shows how much contamination can be expected for a given turbocharger run time. This information is important in determining the feasibility of using an automotive type turbocharger for this application.

# CHAPTER VII

# EXPERIMENTAL RESULTS

The objective of this experiment is to evaluate the performance of a turbocharger based moisture and energy recovery system. This chapter will discuss the experimental results obtained from testing the turbocharger based moisture and energy recovery prototype developed for this investigation. Results include experiments at the nominal design condition with varying mass flow rates. Six experiments were performed in all. Three with humid air and three with dry air. Experiments without steam injection allow an efficiency comparison to determine the effect of high humidity air on turbocharger performance. The turbocharger tested in this investigation features a variable vane type turbine. Variable vane type turbines are capable of changing the angle the fluid strikes the turbine blades to accommodate for many types of flow and loading. In the preliminary testing the vane adjustment is set to the max power setting in an attempt to remove as much energy as possible from the air stream.

## VI.1 Turbocharger Performance Without Steam Injection

This section will discuss the experimental results obtained while running the experiment without steam injection. The air supplied to the system comes directly from an air compressor. The air compressor features a dehumidifier just before the building inlet piping to remove moisture from the compressed air stream. Without steam injection the relative humidity (RH) of the air entering the turbine has been measured and is very low. In most cases the RH is less than 5% and will be neglected to simplify analysis.

The isentropic efficiency is calculated for the dry air case by first determining the isentropic exit temperature ($T_{4s}$) using Equation 7-1.

$$T_{4isen} = T_{3act}\left(\frac{P_{atm}}{P_{3act}}\right)^{(k-1)/k} \tag{7-1}$$

| | | |
|---|---|---|
| Where | $T_{4isen}$ | isentropic turbine exit temperature [C] |
| | $T_{3act}$ | actual turbine inlet temperature [C] |
| | $P_{atm}$ | turbine exhaust pressure [psia] |
| | $P_{3act}$ | actual turbine inlet pressure [psia] |
| | $k$ | specific heat ratio |

The turbine exhausts into the moisture separator, which is assumed for this analysis to have a very low pressure drop. Therefore it is assumed that the turbine exhausts to ambient and therefore $P_{atm}$ is simply the barometric pressure measured in the lab at the time of the experiments. With the isentropic turbine exit temperature known it can be compared with the actual exit temperature to calculate the turbine isentropic efficiency. Equation 7-2 illustrates the equation used in this analysis.

$$\eta_{isen} = \frac{T_{3act} - T_{4act}}{T_{3act} - T_{4isen}} \tag{7-2}$$

| | | |
|---|---|---|
| Where | $T_{4act}$ | actual turbine exit temperature [C] |

A preliminary study on turbocharger dry air efficiency was performed and Figure 7-1 summarizes the results of the experiment.

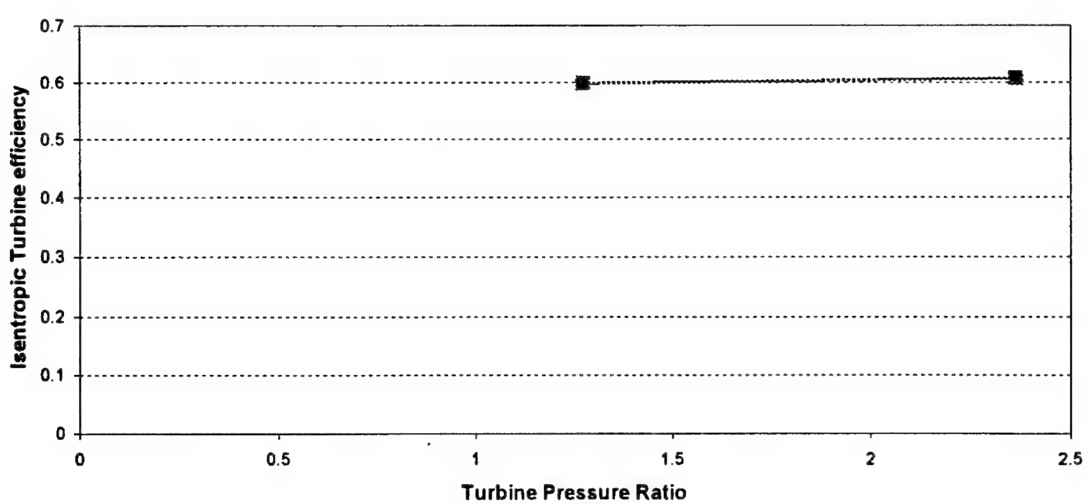


**Figure 7-1 Isentropic turbine efficiency for dry air**

The dry air efficiencies seem to be very stable through the range of pressure ratios encountered in this experiment. An even lower pressure ratio case was run which yielded a very high efficiency. This appears to be caused by experimenter error in taking the exit temperature reading and for this reason is not included in the efficiency analysis. Repeated experimentation to refine measurements is called for to determine more definitely the efficiency for the lower pressure ratios. Figure 7-2 provides flow information at the pressure ratios used in Figure 7-1.

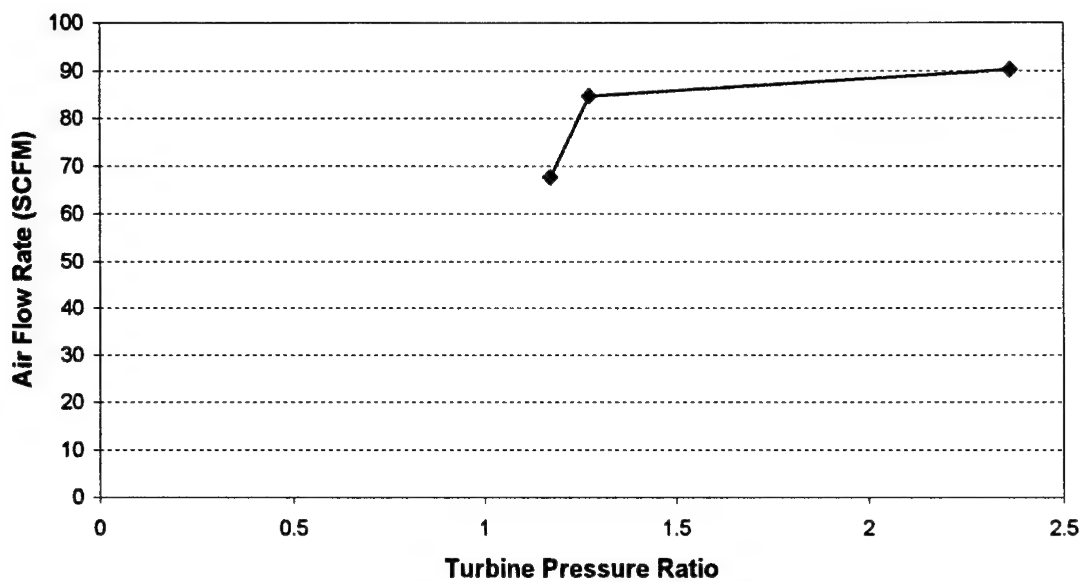


**Figure 7-2 Flow rate versus pressure ratio for dry air.**

The flow initially increases very sharply and then seems to level off. The flow rate information shows that the flow increases with pressure ratio across the turbine.

### VII.2 Humid Air Turbocharger Performance

This section will discuss the experiments performed with steam injection into the turbine air inlet stream. The efficiency analysis on the moist air stream is more complicated than the dry air stream. The air entering and exiting the turbine is considered to be 100% RH. The inlet air ($T_3$)

is assumed to contain only saturated vapor. Equation 7-3 shows the components of the turbine inlet stream

$$h_3 = h_{3air}(T_3) + h_{3vapor}(T_3, P_3) \tag{7-3}$$

Where $h_{3air}$ enthalpy of the turbine inlet dry air

$h_{3vapor}$ enthalpy of the turbine inlet saturated vapor

The exit stream is considered to contain a mixture of dry air, saturated vapor, and liquid droplets. The composition of the exit stream is given in Equation (7-4)

$$h_4 = h_{4air}(T_4) + h_{4vapor}(T_4, P_4) + h_{4liquid}(T_4, P_4) \tag{7-4}$$

Where $h_{4air}$ enthalpy of the turbine exit dry air

$h_{4vapor}$ enthalpy of the turbine exit saturated vapor

$h_{4liquid}$ enthalpy of the turbine exit saturated liquid

Now with the inlet and exhaust stream enthalpies defined these values are used to determine the turbine efficiency running on moist air. The definition of turbine efficiency for the case when moist air testing is performed with condensed liquid in the exhaust stream is provided in equation 7-5.

$$\eta_{isen} = \frac{h_{3sact} - h_{4act}}{h_{3act} - h_{4isen}} \tag{7-5}$$

Where $\eta_{isen}$ moist air turbine isentropic efficiency

$h_{3actual}$ actual enthalpy of the turbine inlet air

$h_{4actual}$ actual enthalpy of the turbine exit air

$h_{4isen}$ isentropic enthalpy of the turbine exit air

Combining Equations 7-4 and 7-5 the calculation of turbine efficiency can be performed. These equations are very difficult to solve by hand and therefore are incorporated into a spreadsheet. This spreadsheet is equipped with a steam table add-in facilitating thermodynamic calculations. The efficiency and percent moisture removal calculations are also performed first in this spreadsheet and then with an Engineering Equation Solver (EES) spreadsheet for comparison. The results agree very well and allow us to proceed with reasonable confidence in these calculations. The programs were used to calculate the turbine efficiencies of the three cases of moist air and the results are summarized in Figure 7-3.

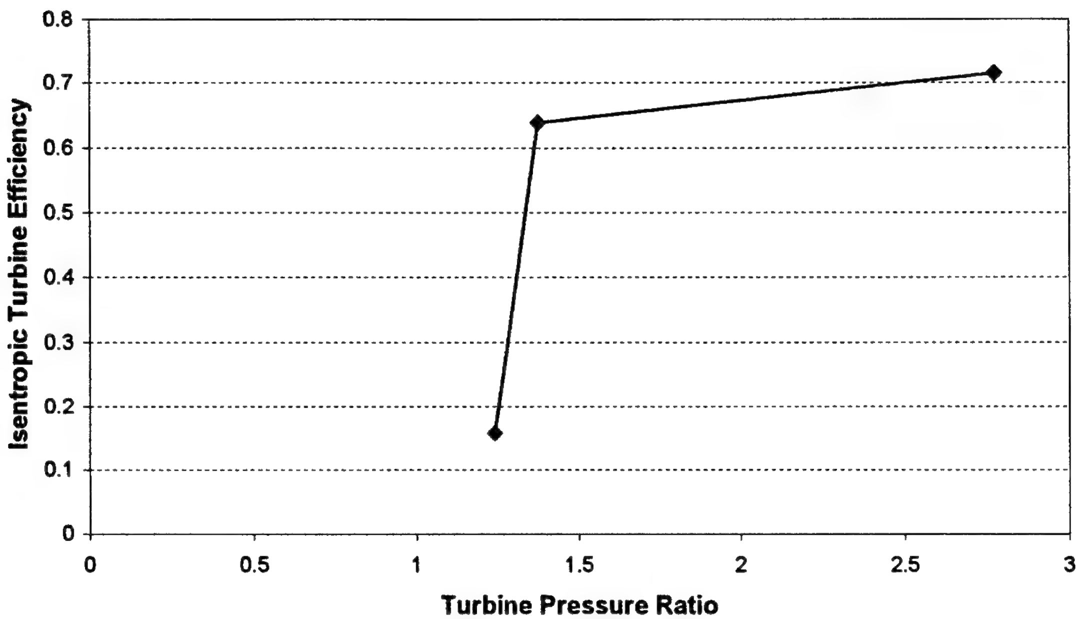


**Figure 7-3 Isentropic turbine efficiency for moist air.**

The efficiency of the turbocharger is relatively high but seems to be very sensitive to pressure ratio. The turbocharger efficiency increases steadily with pressure ratio increase. However, it does appear to level off somewhat after the initial increase. The increased pressure ratio results in increased flow and therefore it appears the turbine could be highly efficient if flows and pressure ratios in excess of these were encountered. Figure 7-4 displays the flow rates corresponding to these pressure ratios for the moist air case. The flow rate increases with pressure drop across the turbine. The initial flow rate increase corresponds to a sharp increase in turbocharger efficiency as would be expected. These results suggest that the low flow dry air

case that yielded a high efficiency was probably in error and that the dry air tests should be repeated and refined to determine the actual turbine efficiencies of low flow rate dry air.

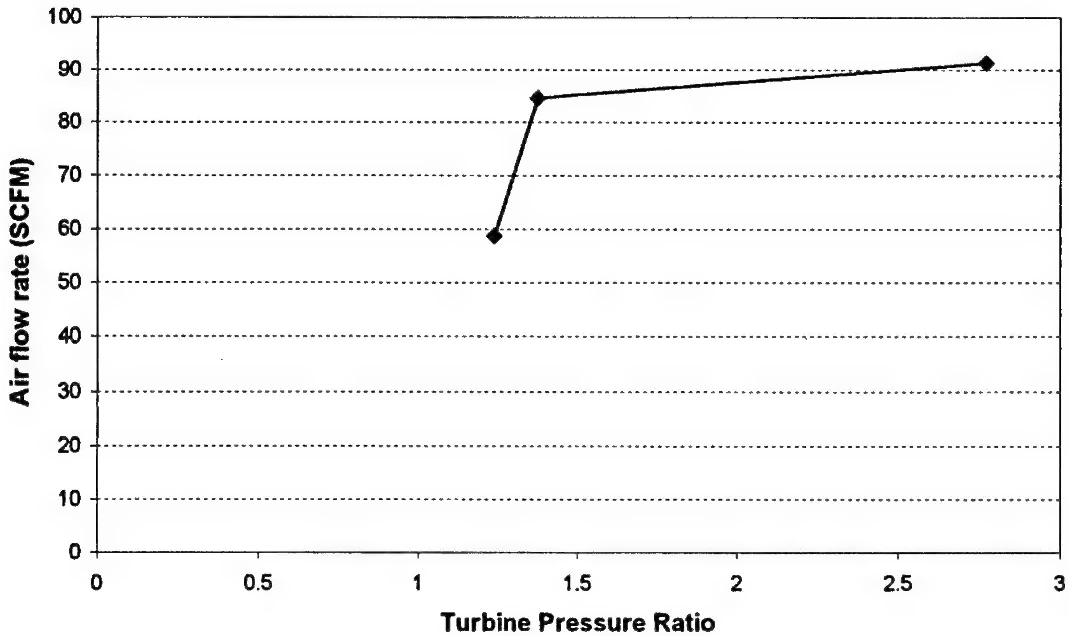


**Figure 7-4 Moist air flow rate versus turbine pressure ratio.**

Moisture removal from the air stream is another parameter examined in this experiment. The aforementioned data analysis spreadsheets calculate the amount of moisture condensed by the turbine. Figure 7-5 shows the calculated percent of the turbine inlet moisture condensing for the three moist air cases run in this experiment. Also included will be the maximum possible moisture that could be removed from the highest pressure ratio case if the turbine behaved isentropically. This will determine if the system is capable of producing enough moisture to facilitate water balance.

The amount of moisture removal is heavily dependant upon pressure ratio. This emphasizes again the need for running the FC at higher pressure. The percent of condensation for the isentropic case at approximately a 3:1 pressure is approximately 23% compared with approximately 17% actually recovered.

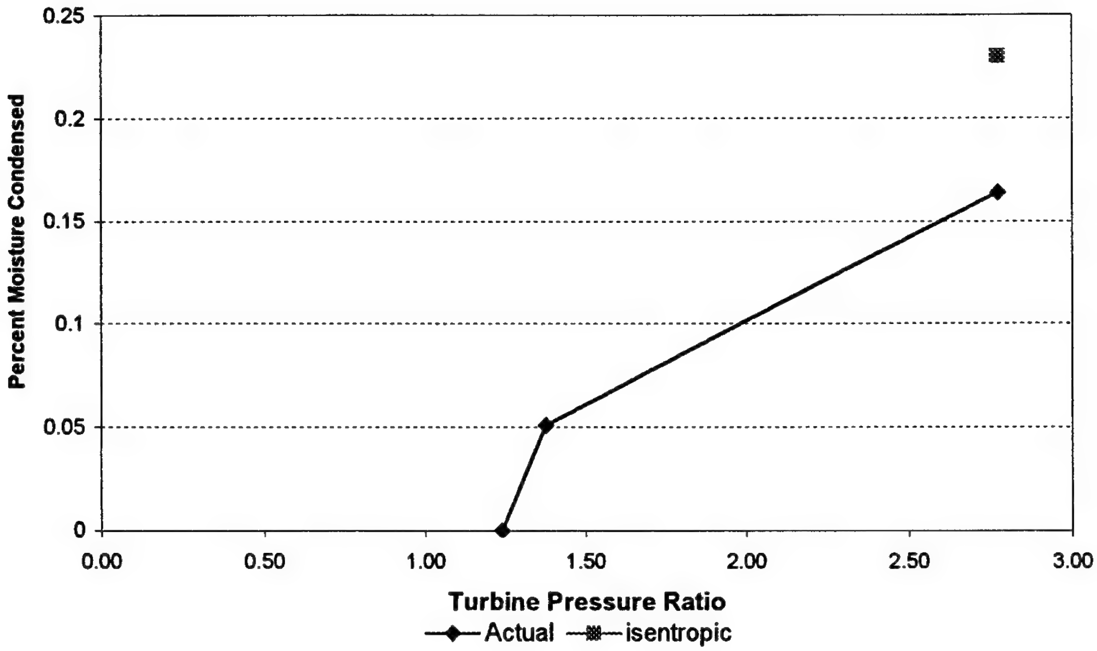


**Figure 7-5 Percent of moisture condensed in turbine exit stream.**

This chapter has displayed an offered a brief discussion of the results. The main findings include an increase in turbocharger efficiency and moisture removal at higher pressures. It is also important to note that for increasing flow rate the turbine efficiency appeared to rise. These findings indicate that the turbocharger system may be able to provide efficient moisture and energy recovery at even higher flow rates and pressures.

# CHAPTER VIII

# EVALUATION

This chapter will discuss the major findings of the experiment and recommend areas of future work to investigate the issues not resolved. The study presented some preliminary results to give an idea of turbocharger performance when used with simulated FC exhaust conditions. The major issue in this investigation is the feasibility of this design for moisture and energy recovery from the exhaust stream of a PEM FC. Many issues will be discussed concerning the system performance and feasibility for FC application.

## VIII.1 Discussion

The moist air turbine efficiency compares very well with the dry air turbine efficiency. The turbine efficiency in both cases is relatively high and allows for energy to be efficiently returned to the FC system in the form of compressed air. The moisture recovery from the turbocharger system is much less than might be expected based on the analysis of dry air operation. Using the present configuration 16% of the turbine inlet moisture could be recovered at approximately 3:1 pressure ratio. Moisture removal for a 3:1 pressure ratio for a turbine operating isentropically was shown to be only 23%. To illustrate the amount of moisture recovery needed to facilitate water balance the chemical equation for the reforming process is given in Equation 8-1. The fuel used in the reforming process is diesel fuel having a standard textbook molecular weight of approximately 170. Therefore $C_{12}H_{26}$ will be used in this analysis.

$$C_{12}\,H_{26} + 24H_2O \rightarrow 12CO_2 + 37H_2 \qquad (8\text{-}1)$$

The 37 moles of H2 can then be burned to generate 37 moles of water vapor. From this balance it can be ascertained that 24 moles must be recovered out of every 37 moles of hydrogen created. Therefore approximately 65% of the moisture must be recovered in order to achieve water balance with no external water sources. The system moisture recovery falls short in the

pressure ranges tested in this phase of the experiment. At an isentropic value of 23% water recovery is still significantly less than the 65% needed for water balance. However, the data shows opportunity for more water removal if the FC is run at higher pressures. It has repeatedly been shown that FCs yield higher efficiencies when operated at higher pressures. The detractor to running at higher pressure is the cost of compressing the inlet air. The turbocharger system reduces that burden, and therefore it may be economically feasible to run at much higher pressure with the turbocharger system in place. In order for this system to be able to supply the required moisture the FC needs to be run at a much higher pressure or design changes to the system need to be made. One possibility being considered for the second phase of the project is the installation of a heat exchanger following the turbine to further reduce the temperature and increase condensation. The added heat exchanger would still allow for energy removal from the system and would enhance the moisture removal. This heat exchanger might even be some innovative form of cooled separator that could both remove heat and separate condensed moisture.

Another topic to be discusses is oil contamination from the moist air stream. This problem was discovered late in the first phase of this project and only one test was performed. The test results showed after approximately four hours of moist air run time the oil being supplied to the turbocharger was approximately 20% water by weight and 19% by volume. This test may be cause for some concern in this experiment as they have been in previous similar experiments such as those performed by Grabow (1986) and Mctaggert (1998). However, the main concern is compressor outlet air contamination. As stated previously, the compressor outlet air will ultimately be used as combustion air in the combustion process. To monitor compressor outlet air quality, immediately after the compressor outlet is installed a clear plastic hose. No oil residue or discoloration has been observed in this hose thus far. If no compressor outlet contamination occurs the only oiling issue is with the life of the turbine. The effect on the turbocharger resulting from the water content of the oil has not yet been observed. The turbocharger has been operated with moist inlet air for more that 25 hours with no problems. The oil contamination problem was recently discovered and therefore limited information on the extent or affect oil contamination will have on the life of the turbine is not yet known. However,

the oil water mixture seems to still maintain lubrication properties and for rugged automotive type turbochargers may not pose a problem.

The issue of damage to turbine blades from the water droplets separating out of the moist air stream is another issue to be explored. Turbine damage due to water droplets does not appear to be a problem at this stage in the investigation. It is even possible that the majority of the condensation is occurring outside of the turbocharger. Recently the turbocharger was dismantled exposing the turbine blades. Inspection of the blades showed no sign of unusual wear that could be attributed to liquid particle striking the turbine blades. However, attempts will be made to increase moisture removal efficiency resulting in some changes to the system. One such attempt will be made in the near future to increase the condensation of moisture in the turbine. If greater condensation becomes reality, turbine blade damage may become an issue.

In summary, the turbocharger moisture recovery is much less than expected. Without modification to the system itself or a change in the FC operating pressure, the system will not be able to fully supply the water for the FC reforming system. This is a disappointment, but at such high turbine efficiencies much energy is returned to the system improving the overall system efficiency. The topics of turbocharger oil contamination and turbine blade damage from condensing droplets will have to be examined in more detail in future experiments but are not major problems now. It is important to remember that this is a preliminary attempt at turbocharger type systems for moisture and energy recovery from FCs, and many options remain to be investigated.

### VIII.2 Recommendations

The initial phase of the FC moisture and energy recovery system is complete. The objective of the project was to design, test, and determine the feasibility of a moisture and energy recovery system for FC exhaust. In order to determine the feasibility of the system testing at the nominal design condition was performed. In addition, testing of the system was performed at varying flow rates to simulate part load conditions frequently encountered in FCs.

The future work will be performed in the second phase of the project and will include an investigation into alternative system designs to remove more water from the exhaust stream. The inability of the system to remove the required amount of moisture is the only shortcoming of the system, and an alternative should be developed. Once a final system is decided upon, a detailed parametric study of the system should be performed. The moisture and energy removal system inlet conditions will be varied based on the parameters of flow rate, temperature, pressure, and relative humidity. The effects of these parameters on the turbocharger performance will be determined allowing for possible optimization of the system. This optimization should include examining the FC system to determine what FC operating conditions are most efficient if the prototype moisture and energy recovery system designed in this experiment is considered in the analysis. Work should also be performed to measure the RPM of the turbocharger and compare this data to published turbocharger manufacturer data. This research will show what turbocharger performance can be expected at different speeds.

More data needs be gathered on oil and air contamination to determine the feasibility of an automotive turbocharger system for FC application. The effect on the turbocharger performance and life needs to be determined. The water content of the oil supplied to the turbocharger needs to be continuously monitored in the second phase of testing. This data will illustrate the extent of the contamination and allow for the prediction of when oil change outs should occur. If this type of turbocharger turns out to not be feasible, turbocharger technology using air bearings is currently available and should be further researched in future works on this project.

In conclusion, the future works on this project should focus on initially re-designing the moisture and energy recovery system to enhance moisture removal. Once a system is decided upon extensive testing should be performed on all aspects of this system. The collection of performance data should allow a determination of under what conditions the system performs the desired level of moisture and energy recovery.

## APPENDIX A

## RAW DATA

Date November 3, 2001

Barometer Reading 14.678 psig

Standard Absolute Temperature for F1_meas = 294.44 K

Standard Absolute Temperature for F9_meas = 298.00 K

Standard Absolute Pressure for F1_meas = 46.7 psia

Standard Absolute Pressure for F9_meas = 14.7 psia

| | T1 | T2 | T3 | T4 | T5 | T6 | T7 | T8 |
|---|---|---|---|---|---|---|---|---|
| | C | C | C | C | C | C | C | C |
| Dry Air | 20.71 | 63.48 | 60.25 | 16.09 | 17.70 | 38.20 | 19.91 | 19.18 |
| Moist Air | 20.85 | 24.26 | 80.66 | 54.02 | 54.15 | 58.57 | 133.16 | 20.96 |

| | T9 | P1 | P1 | F1_mea | F1_corr | P3 | P3 |
|---|---|---|---|---|---|---|---|
| | C | psig | psia | SCFM | SCFM | psig | psia |
| Dry Air | 85.56 | 44.5 | 59.2 | 80.0 | 90.2 | 20.0 | 34.7 |
| Moist Air | 111.42 | 46.0 | 60.7 | 80.0 | 91.3 | 26.0 | 40.7 |

| | P9 | P9 | F9_meas | F9_meas | F9_corr | RH6 | Sep flow |
|---|---|---|---|---|---|---|---|
| | psig | psia | % FS | SCFM | SCFM | % | mL/min |
| Dry Air | 8.6 | 23.3 | 43 | 32.3 | 37.0 | 4.31 | 0 |
| Moist Air | 12.4 | 27.1 | 49 | 36.8 | 43.9 | 97.64 | 100 |

Date November 3, 2001

Barometer Reading 14.678 psig

Standard Absolute Temperature for F1_meas = 294.44 K

Standard Absolute Temperature for F9_meas = 298.00 K

Standard Absolute Pressure for F1_meas = 46.7 psia

Standard Absolute Pressure for F9_meas = 14.7 psia

| | T1 | T2 | T3 | T4 | T5 | T6 | T7 | T8 |
|---|---|---|---|---|---|---|---|---|
| | C | C | C | C | C | C | C | C |
| Dry Air | 19.35 | 63.09 | 59.97 | 46.64 | 46.05 | 28.90 | 40.33 | 20.68 |
| Moist Air | 19.31 | 58.53 | 60.80 | 53.13 | 52.85 | 48.42 | 111.72 | 21.50 |

| | T9 | P1 | P1 | F1_mea | F1_corr | P3 | P3 |
|---|---|---|---|---|---|---|---|
| | C | psig | psia | SCFM | SCFM | psig | psia |
| Dry Air | 40.84 | 32.0 | 46.7 | 80.0 | 80.3 | 4.0 | 18.7 |
| Moist Air | 49.40 | 32.5 | 47.2 | 84.0 | 84.7 | 5.5 | 20.2 |

| | P9 | P9 | F9_meas | F9_meas | F9_corr | RH6 | Sep flow |
|---|---|---|---|---|---|---|---|
| | psig | psia | % FS | SCFM | SCFM | % | mL/min |
| Dry Air | 1.8 | 16.5 | 21 | 15.8 | 16.2 | 5.31 | 0.0 |
| Moist Air | 2.6 | 17.3 | 29 | 21.8 | 22.7 | 101.48 | 50.0 |

Date November 3, 2001

Barometer Reading 14.678 psig

Standard Absolute Temperature for F1_meas = 294.44 K

Standard Absolute Temperature for F9_meas = 298.00 K

Standard Absolute Pressure for F1_meas = 46.7 psia

Standard Absolute Pressure for F9_meas = 14.7 psia

| | T1 | T2 | T3 | T4 | T5 | T6 | T7 | T8 |
|---|---|---|---|---|---|---|---|---|
| | C | C | C | C | C | C | C | C |
| Dry Air | 19.31 | 64.31 | 60.42 | 49.48 | 47.31 | 26.79 | 51.55 | 20.21 |
| Moist Air | 19.16 | 23.54 | 69.77 | 65.53 | 65.63 | 59.24 | 109.71 | 21.15 |

| | T9 | P1 | P1 | F1_mea | F1_corr | P3 | P3 |
|---|---|---|---|---|---|---|---|
| | C | psig | psia | SCFM | SCFM | psig | psia |
| Dry Air | 36.88 | 23.0 | 37.7 | 75.0 | 67.6 | 2.5 | 17.2 |
| Moist Air | 37.34 | 20.0 | 34.7 | 68.0 | 58.8 | 3.5 | 18.2 |

| | P9 | P9 | F9_meas | F9_meas | F9_corr | RH6 | Sep flow |
|---|---|---|---|---|---|---|---|
| | psig | psia | % FS | SCFM | SCFM | % | mL/min |
| Dry Air | 1.2 | 15.9 | 16 | 12.0 | 12.2 | 6.48 | 0.0 |
| Moist Air | 1.5 | 16.2 | 21 | 15.8 | 16.2 | 95.42 | 25.0 |

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